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Kikuchi et al.

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(45) **Date of Patent: Jul. 7, 2015**

(54) **VEHICLE CONTROL DEVICE**

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(73) Assignee: **Nissan Motor Co., Ltd.**, Yokohama (JP)

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PCT Pub. Date: **Aug. 1, 2013**

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(30) **Foreign Application Priority Data**

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B60G 17/06 (2006.01)
B60G 17/0165 (2006.01)
(Continued)

(52) **U.S. Cl.**
CPC ***B60G 17/0165*** (2013.01); ***B60G 17/0182*** (2013.01); ***B60G 17/08*** (2013.01);
(Continued)

(58) **Field of Classification Search**
CPC B60G 17/06; B60G 17/018; B60G 17/015; B60W 10/22
USPC 701/37, 38; 280/5.502
See application file for complete search history.

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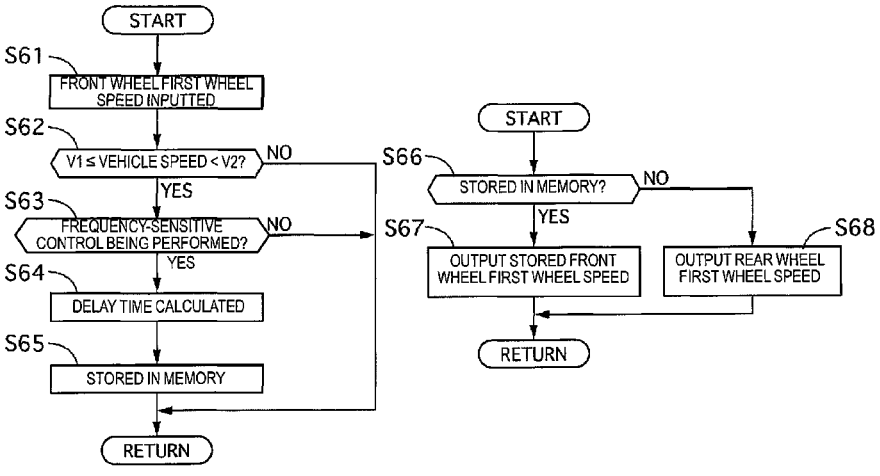
Primary Examiner — Tan Q Nguyen
(74) *Attorney, Agent, or Firm* — Global IP Counselors, LLP

(57) **ABSTRACT**

A vehicle control device includes a variable-damping-force shock absorber, a front wheel speed sensor, a rear wheel speed estimation device, a damping force control amount calculation device configured to calculate a damping force control amount for the variable-damping-force shock absorber according to frequency scalar quantities, when the front wheel speed detected by the front wheel speed sensors and the size of an amplitude of a frequency of the rear wheel speed estimated by the rear wheel speed estimation device are frequency scalar quantities, and a damping force control device configured to control the damping force of the variable-damping-force shock absorber on the basis of the damping force control amount calculated by the damping force control amount calculation device, the rear wheel speed estimation device prohibiting the estimation of the real wheel speed when a vehicle speed is equal to or greater than a high vehicle speed threshold value.

13 Claims, 15 Drawing Sheets



(51) **Int. Cl.**

B60G 17/018	(2006.01)
B60G 17/08	(2006.01)
B60G 17/0195	(2006.01)
B60W 10/06	(2006.01)
B60W 10/18	(2012.01)
B60W 10/22	(2006.01)

(52) **U.S. Cl.**

CPC *B60G2400/204* (2013.01); *B60G 2400/208* (2013.01); *B60G 2400/91* (2013.01); *B60G 2500/106* (2013.01); *B60G 2600/188* (2013.01); *B60G 2800/012* (2013.01); *B60G 2800/014* (2013.01); *B60G 2800/162* (2013.01); *B60G 2800/916* (2013.01); *B60G 2500/10* (2013.01); ***B60G 17/0195*** (2013.01); ***B60W 10/06*** (2013.01); ***B60W 10/18*** (2013.01); ***B60W 10/22*** (2013.01); *B60G 2800/702* (2013.01)

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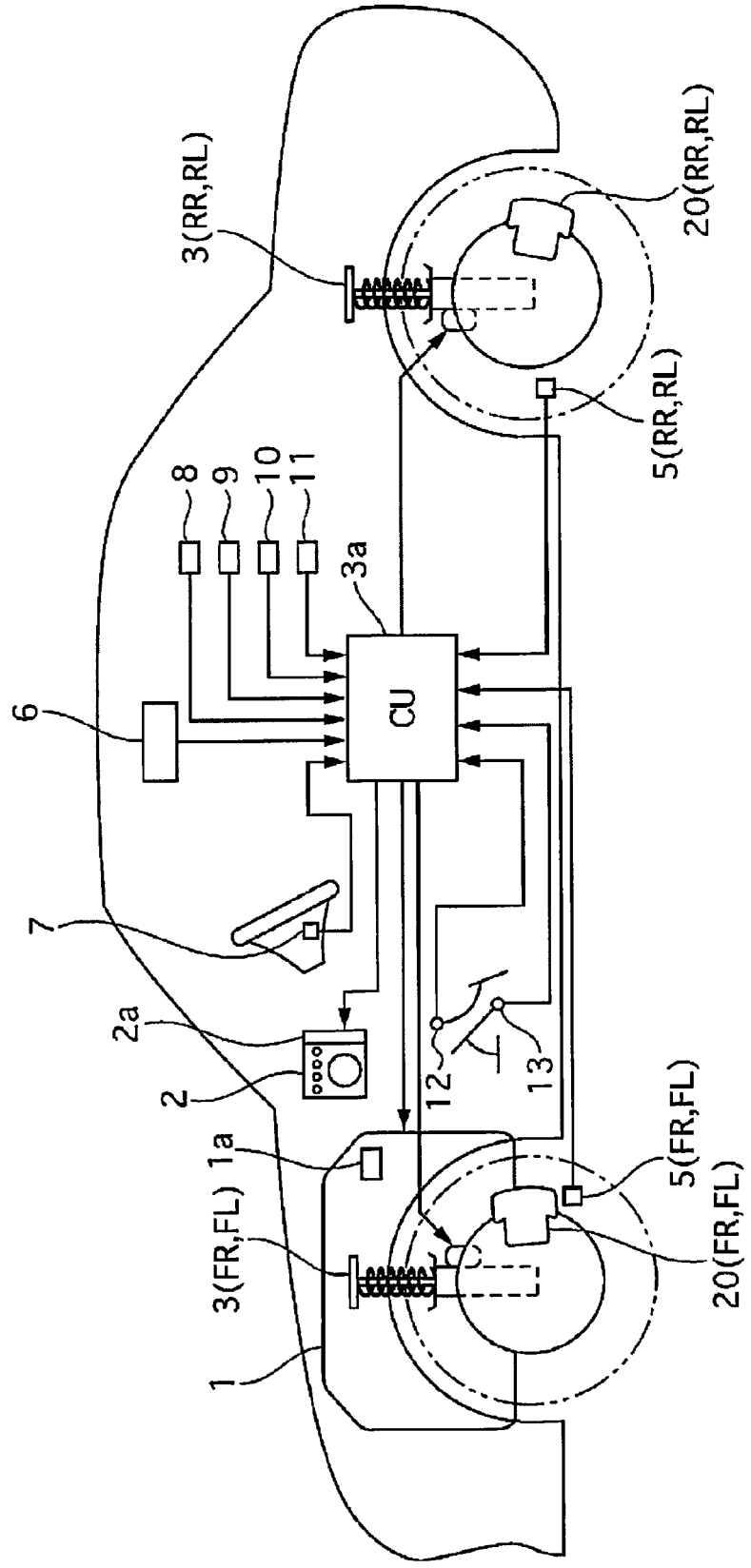
3(RR,RL)
20(RR,RL)
5(RR,RL)
8
9
10
11
3a
CU
6
7
2a
2
12
13
1a
3(FR,FL)
5(FR,FL)
20(FR,FL)
1

FIG. 1

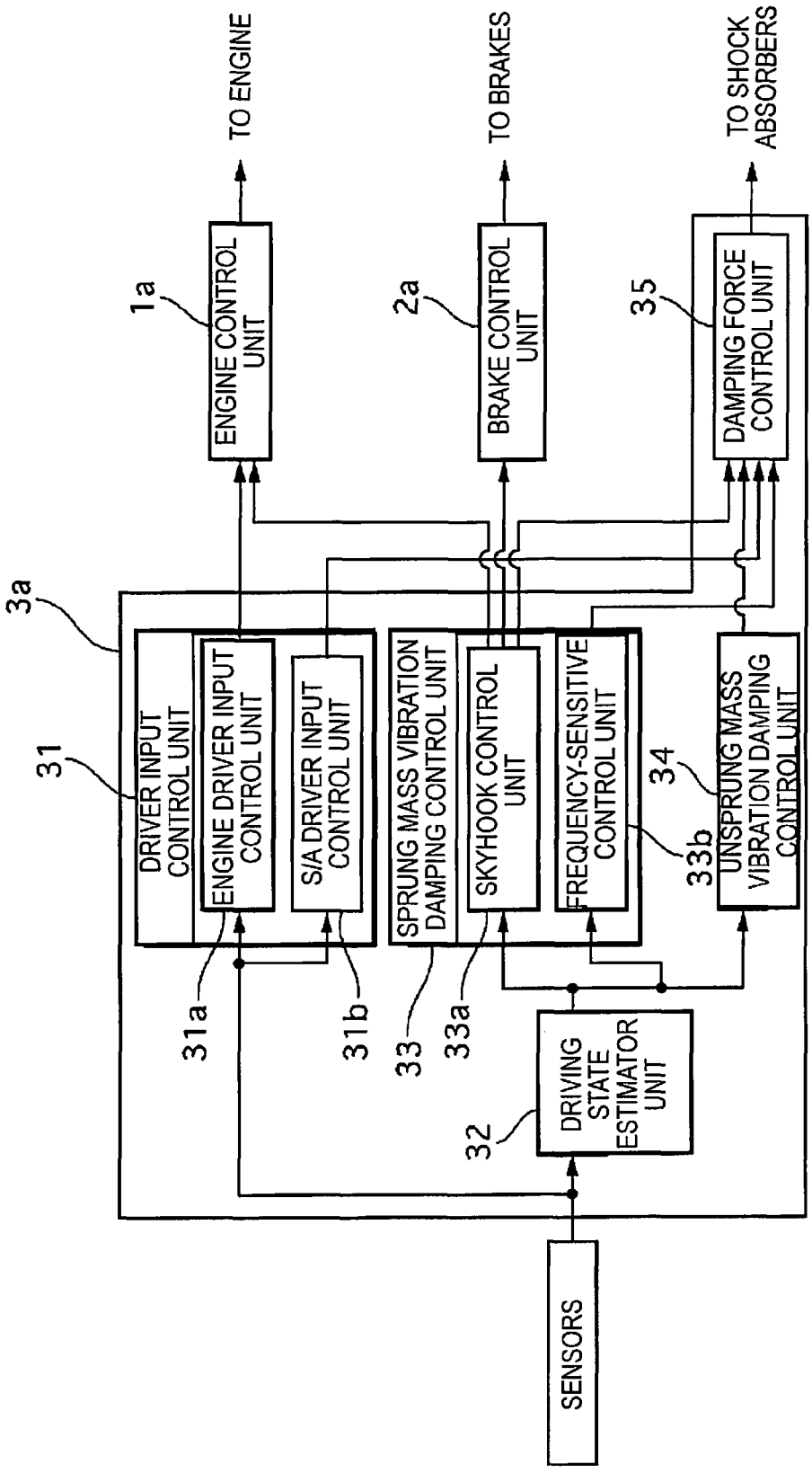
TO ENGINE
TO BRAKES
TO SHOCK ABSORBERS
1a
ENGINE CONTROL UNIT
2a
BRAKE CONTROL UNIT
35
DAMPING FORCE CONTROL UNIT
3a
31
DRIVER INPUT CONTROL UNIT
ENGINE DRIVER INPUT CONTROL UNIT
S/A DRIVER INPUT CONTROL UNIT
SPRUNG MASS VIBRATION DAMPING CONTROL UNIT
SKYHOOK CONTROL UNIT
FREQUENCY-SENSITIVE CONTROL UNIT
34
33b
UNSPRUNG MASS VIBRATION DAMPING CONTROL UNIT
31a
31b
33
33a
32
DRIVING STATE ESTIMATOR UNIT
SENSORS

FIG. 2

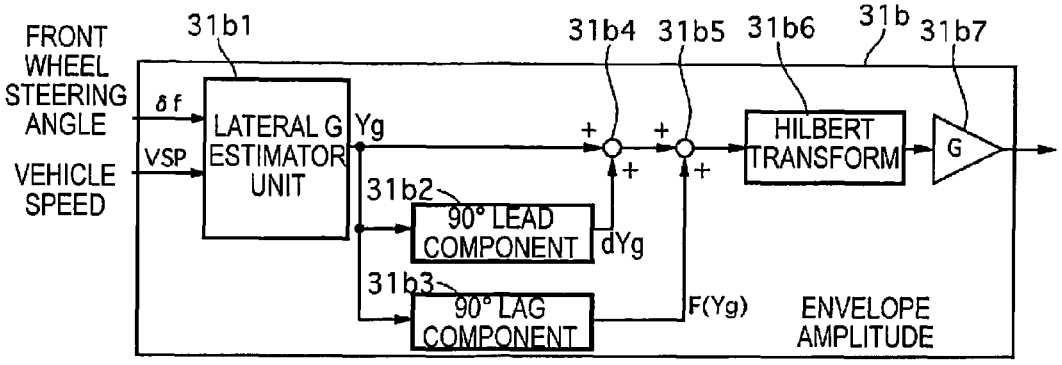

FIG. 3

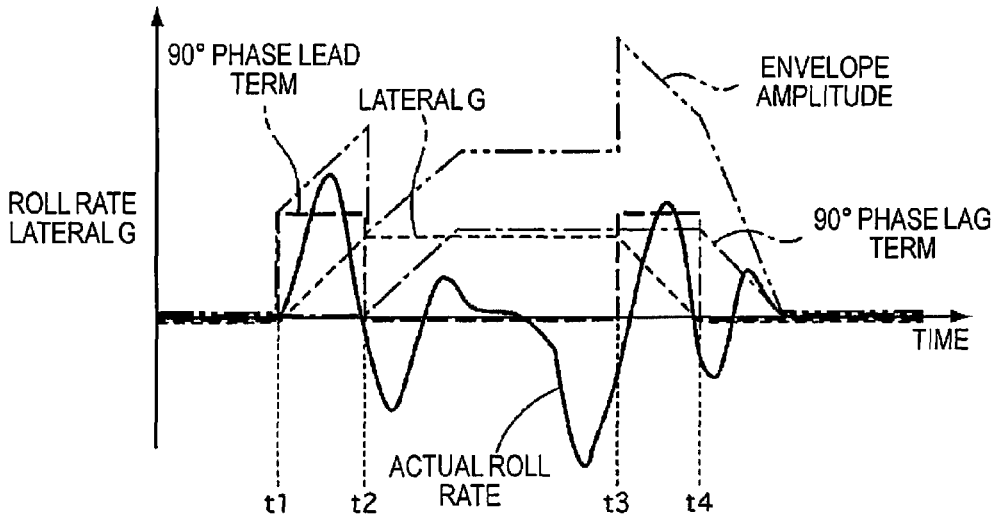

FIG. 4

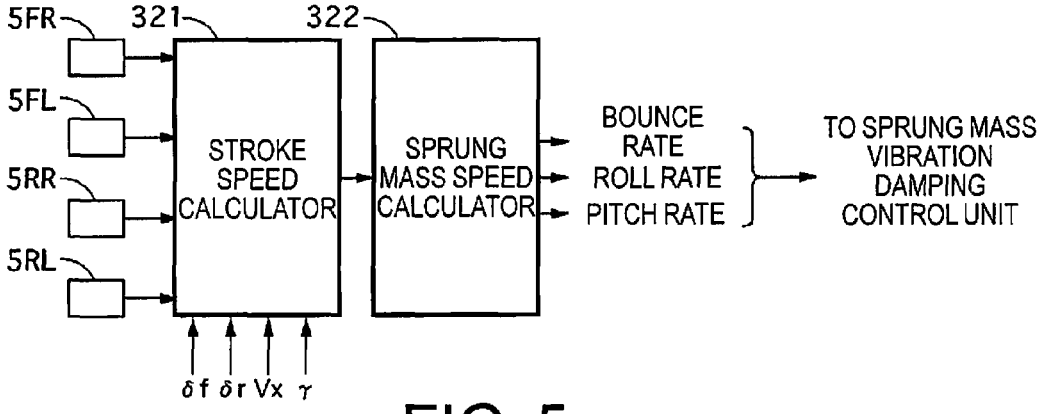

FIG. 5

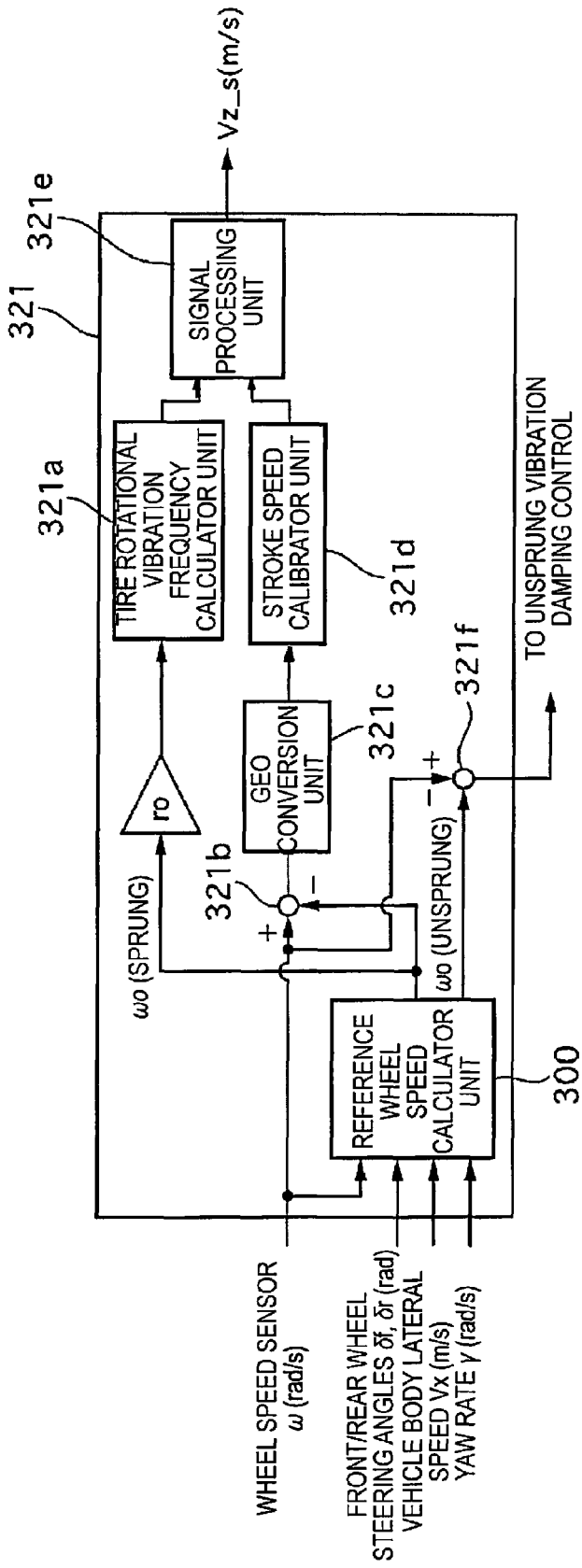
Vz_s(m/s)
321e
321
SIGNAL PROCESSING UNIT
321a
TIRE ROTATIONAL VIBRATION FREQUENCY CALCULATOR UNIT
STROKE SPEED CALIBRATOR UNIT
321d
TO UNSPRUNG VIBRATION DAMPING CONTROL
321f
321c
GEO CONVERSION UNIT
ro
321b
ωo (SPRUNG)
ωo (UNSPRUNG)
REFERENCE WHEEL SPEED CALCULATOR UNIT
300
WHEEL SPEED SENSOR
ω (rad/s)
FRONT/REAR WHEEL STEERING ANGLES δf, δr (rad)
VEHICLE BODY LATERAL SPEED Vx (m/s)
YAW RATE γ (rad/s)

FIG. 6

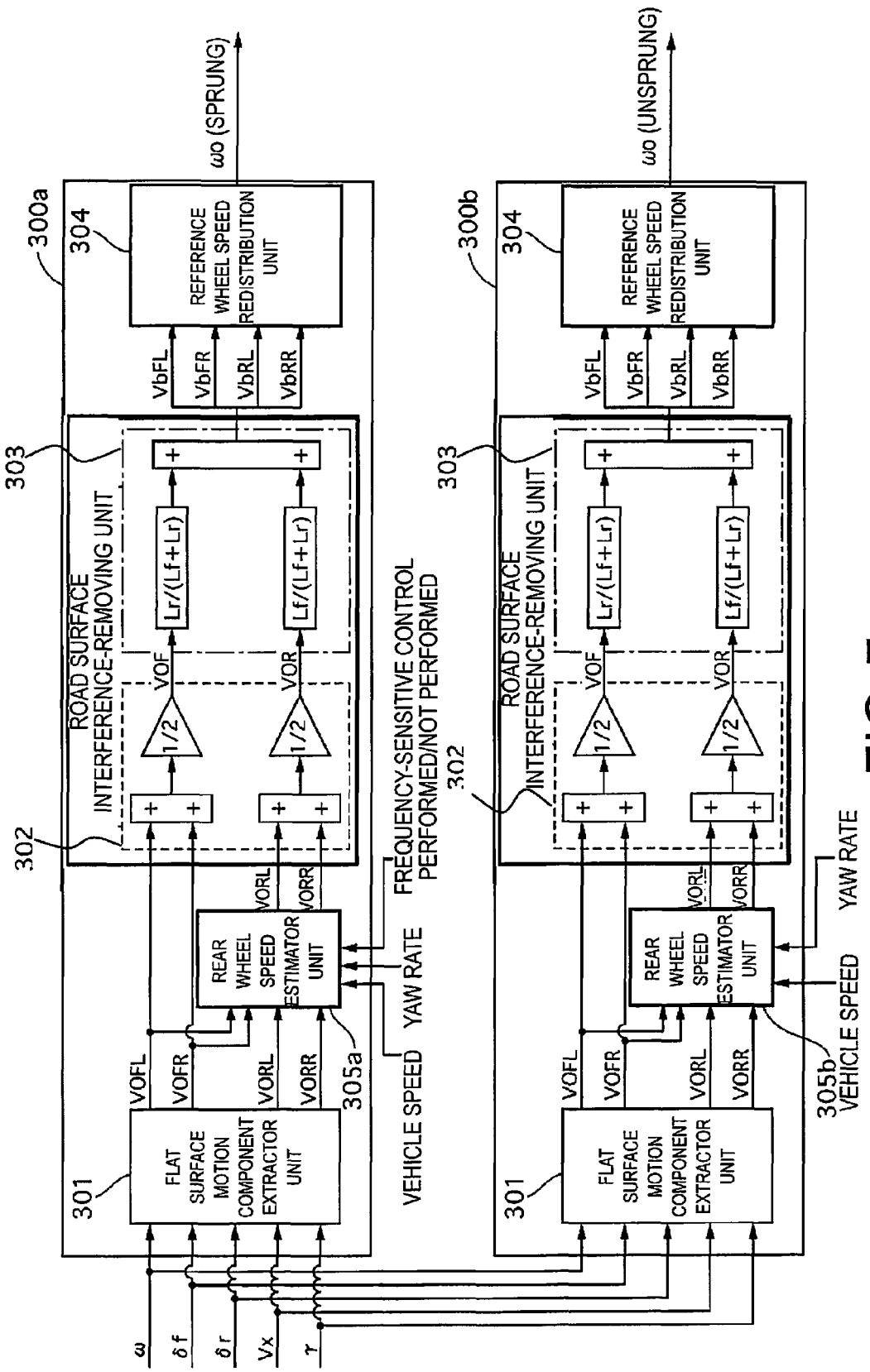
300a
300b
303
304
302
301
ωo (SPRUNG)
ωo (UNSPRUNG)
REFERENCE WHEEL SPEED REDISTRIBUTION UNIT
ROAD SURFACE INTERFERENCE-REMOVING UNIT
FLAT SURFACE MOTION COMPONENT EXTRACTOR UNIT
REAR WHEEL SPEED ESTIMATOR UNIT
VbFL
VbFR
VbRL
VbRR
VOF
VOR
VOFL
VOFR
VORL
VORR
Lr/(Lf+Lr)
Lf/(Lf+Lr)
1/2
FREQUENCY-SENSITIVE CONTROL PERFORMED/NOT PERFORMED
YAW RATE
VEHICLE SPEED
305a
305b
ω
δf
δr
Vx
τ

FIG. 7

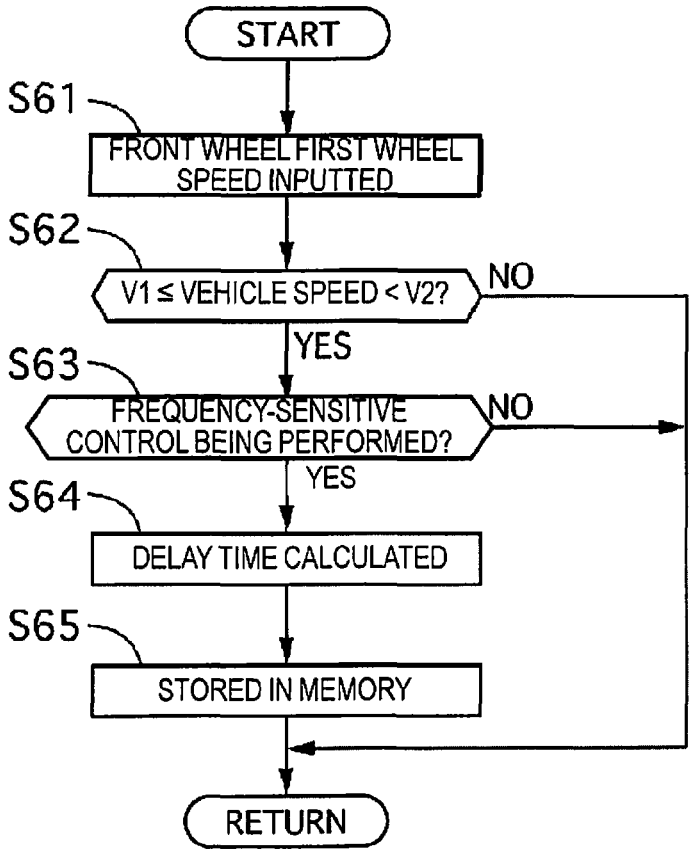
START
S61
FRONT WHEEL FIRST WHEEL SPEED INPUTTED
S62
V1 ≤ VEHICLE SPEED < V2?
NO
YES
S63
FREQUENCY-SENSITIVE CONTROL BEING PERFORMED?
NO
YES
S64
DELAY TIME CALCULATED
S65
STORED IN MEMORY
RETURN

FIG. 8A

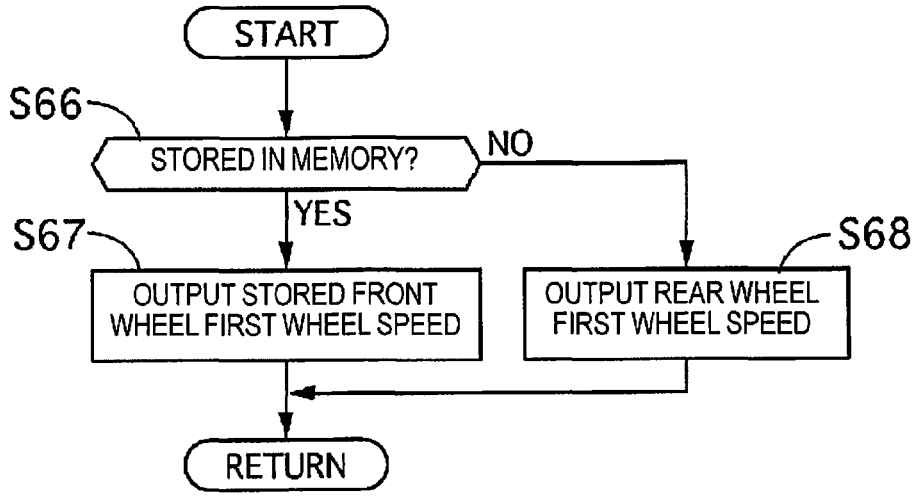
START
S66
STORED IN MEMORY?
NO
YES
S67
OUTPUT STORED FRONT WHEEL FIRST WHEEL SPEED
S68
OUTPUT REAR WHEEL FIRST WHEEL SPEED
RETURN

FIG. 8B

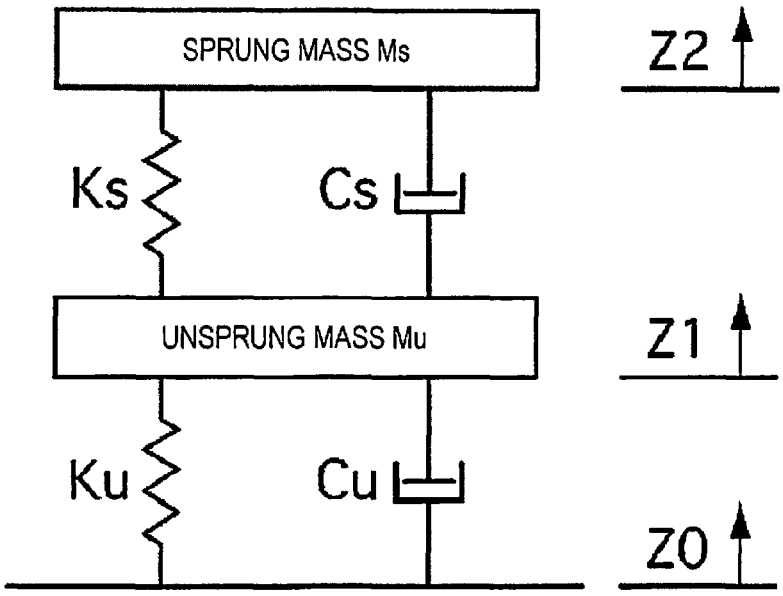
SPRUNG MASS Ms
Z2
Ks
Cs
UNSPRUNG MASS Mu
Z1
Ku
Cu
Z0

FIG. 9A

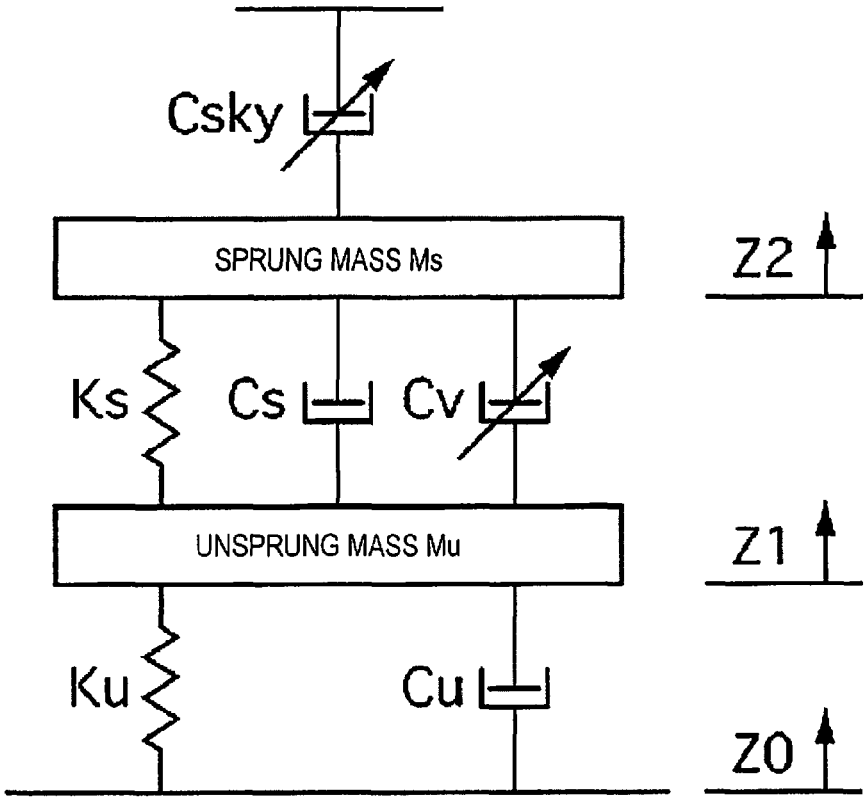
Csky
SPRUNG MASS Ms
Z2
Ks
Cs
Cv
UNSPRUNG MASS Mu
Z1
Ku
Cu
Z0

FIG. 9B

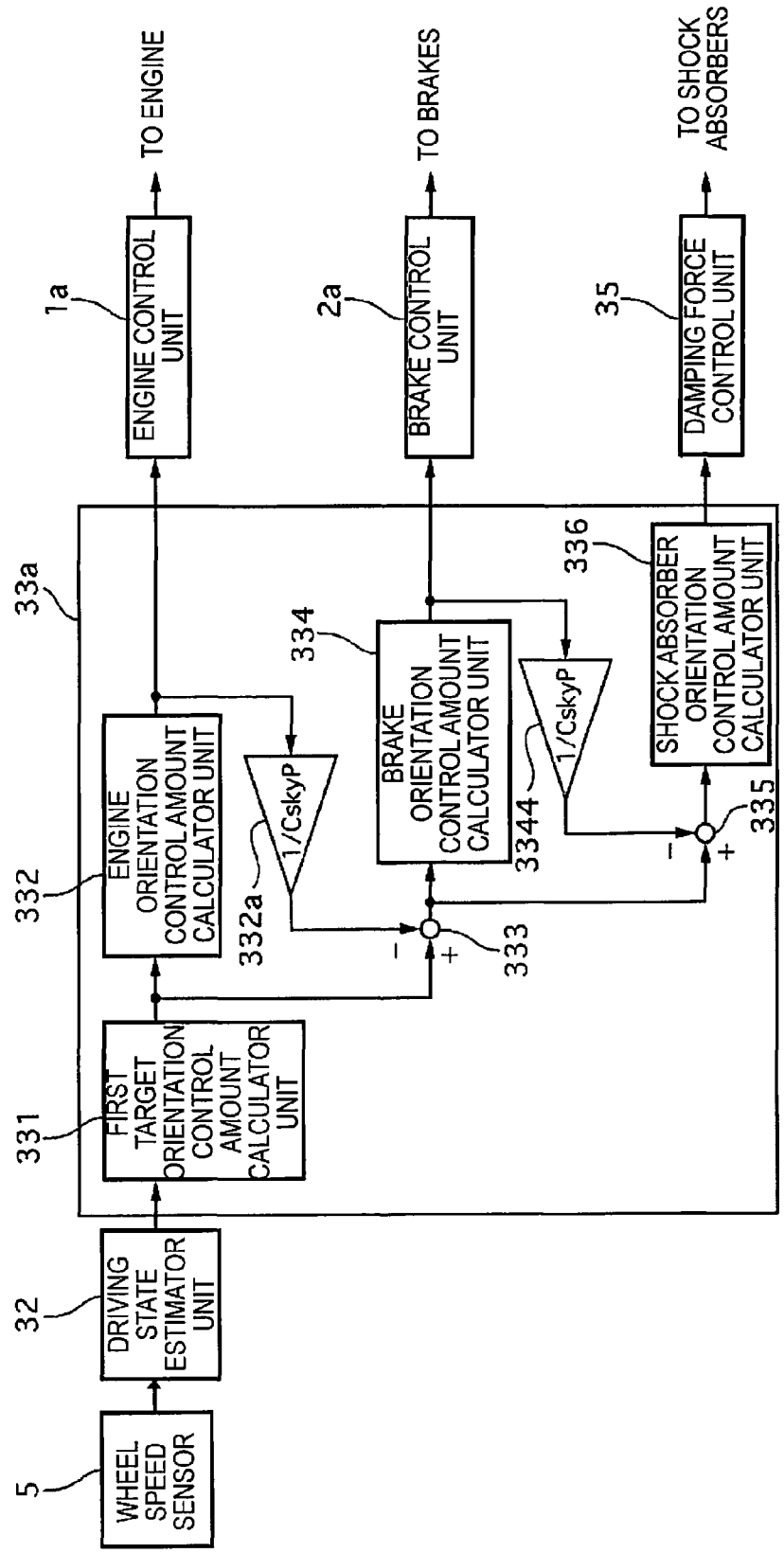
5
WHEEL SPEED SENSOR
32
DRIVING STATE ESTIMATOR UNIT
331
FIRST TARGET ORIENTATION CONTROL AMOUNT CALCULATOR UNIT
332
ENGINE ORIENTATION CONTROL AMOUNT CALCULATOR UNIT
332a
1/CskyP
333
334
BRAKE ORIENTATION CONTROL AMOUNT CALCULATOR UNIT
3344
1/CskyP
335
336
SHOCK ABSORBER ORIENTATION CONTROL AMOUNT CALCULATOR UNIT
33a
1a
ENGINE CONTROL UNIT
TO ENGINE
2a
BRAKE CONTROL UNIT
TO BRAKES
35
DAMPING FORCE CONTROL UNIT
TO SHOCK ABSORBERS

FIG. 10

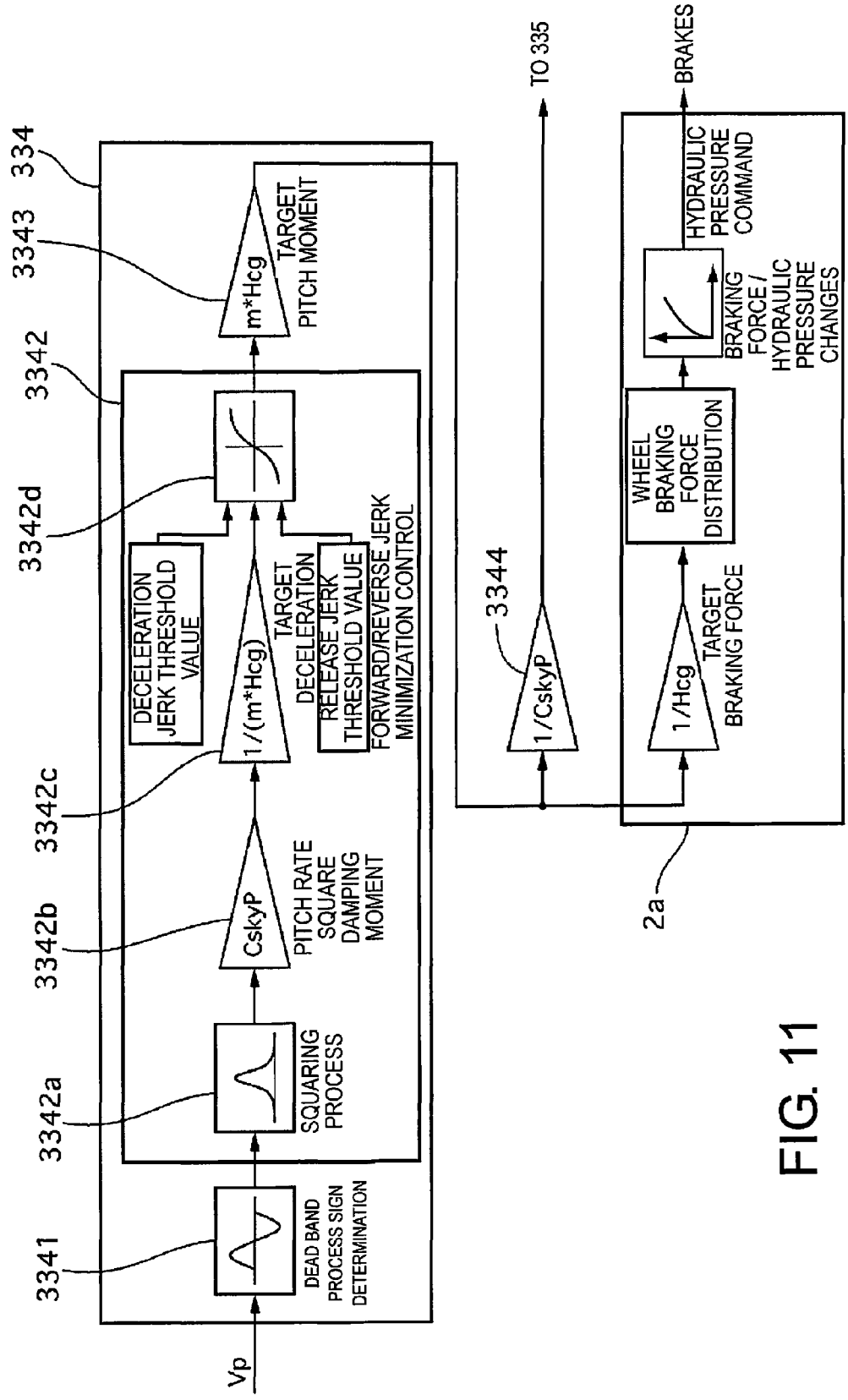
Vp
3341
DEAD BAND PROCESS SIGN DETERMINATION
3342a
SQUARING PROCESS
3342b
CskyP
PITCH RATE SQUARE DAMPING MOMENT
3342c
1/(m*Hcg)
TARGET DECELERATION
DECELERATION JERK THRESHOLD VALUE
RELEASE JERK THRESHOLD VALUE
FORWARD/REVERSE JERK MINIMIZATION CONTROL
3342d
3342
3343
m*Hcg
TARGET PITCH MOMENT
334
TO 335
3344
1/CskyP
2a
1/Hcg
TARGET BRAKING FORCE
WHEEL BRAKING FORCE DISTRIBUTION
BRAKING FORCE / HYDRAULIC PRESSURE CHANGES
HYDRAULIC PRESSURE COMMAND
BRAKES

FIG. 11

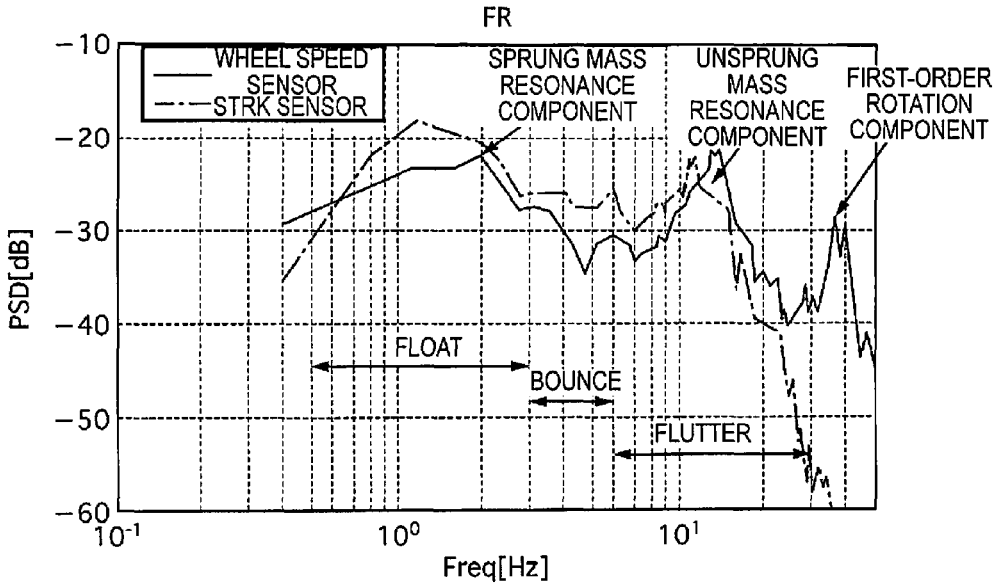
FR
WHEEL SPEED SENSOR
STRK SENSOR
SPRUNG MASS RESONANCE COMPONENT
UNSPRUNG MASS RESONANCE COMPONENT
FIRST-ORDER ROTATION COMPONENT
FLOAT
BOUNCE
FLUTTER
PSD[dB]
−10
−20
−30
−40
−50
−60
10^-1
10^0
10^1
Freq[Hz]

FIG. 12

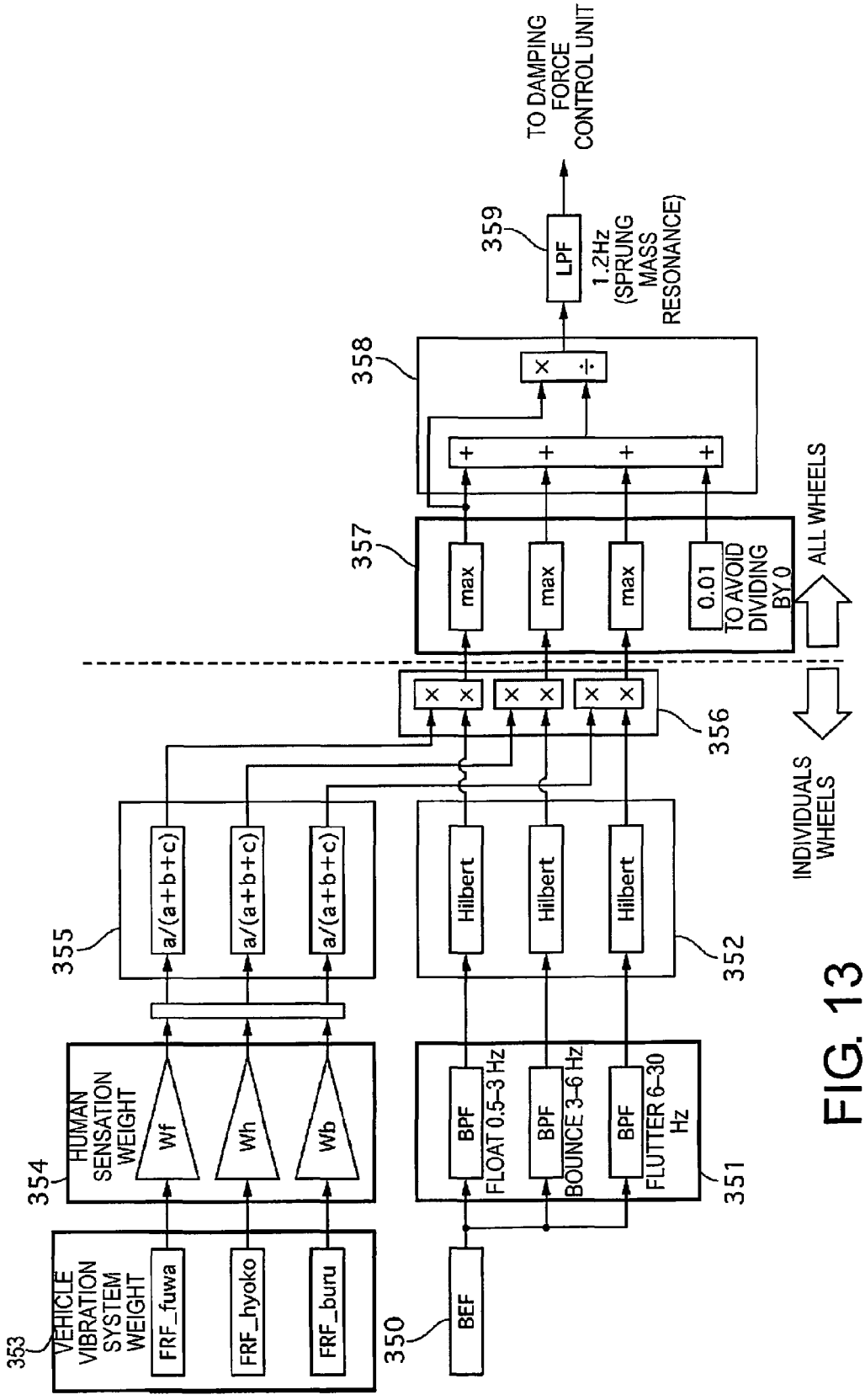
353
VEHICLE VIBRATION SYSTEM WEIGHT
FRF_fuwa
FRF_hyoko
FRF_buru
350
BEF
354
HUMAN SENSATION WEIGHT
Wf
Wh
Wb
355
a/(a+b+c)
a/(a+b+c)
a/(a+b+c)
351
BPF
FLOAT 0.5–3 Hz
BPF
BOUNCE 3–6 Hz
BPF
FLUTTER 6–30 Hz
352
Hilbert
Hilbert
Hilbert
INDIVIDUALS WHEELS
356
357
max
max
max
0.01
TO AVOID DIVIDING BY 0
ALL WHEELS
358
359
LPF
1.2Hz (SPRUNG MASS RESONANCE)
TO DAMPING FORCE CONTROL UNIT

FIG. 13

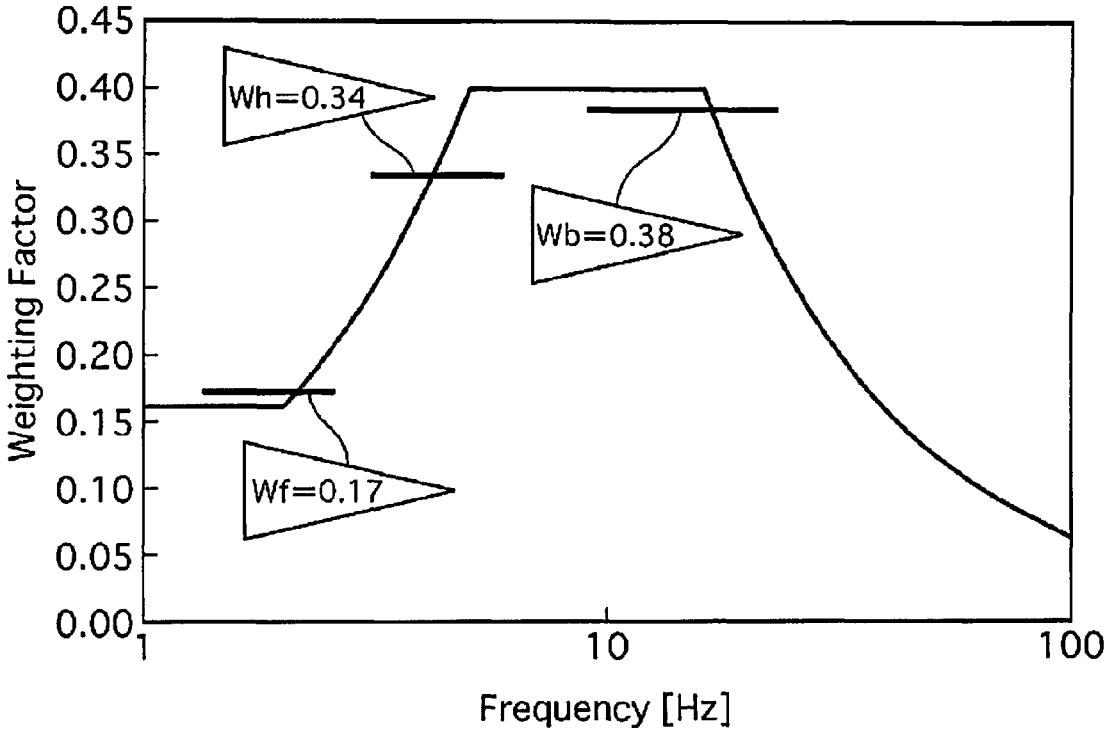
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0.00
Weighting Factor
Wh=0.34
Wb=0.38
Wf=0.17
1
10
100
Frequency [Hz]

FIG. 14

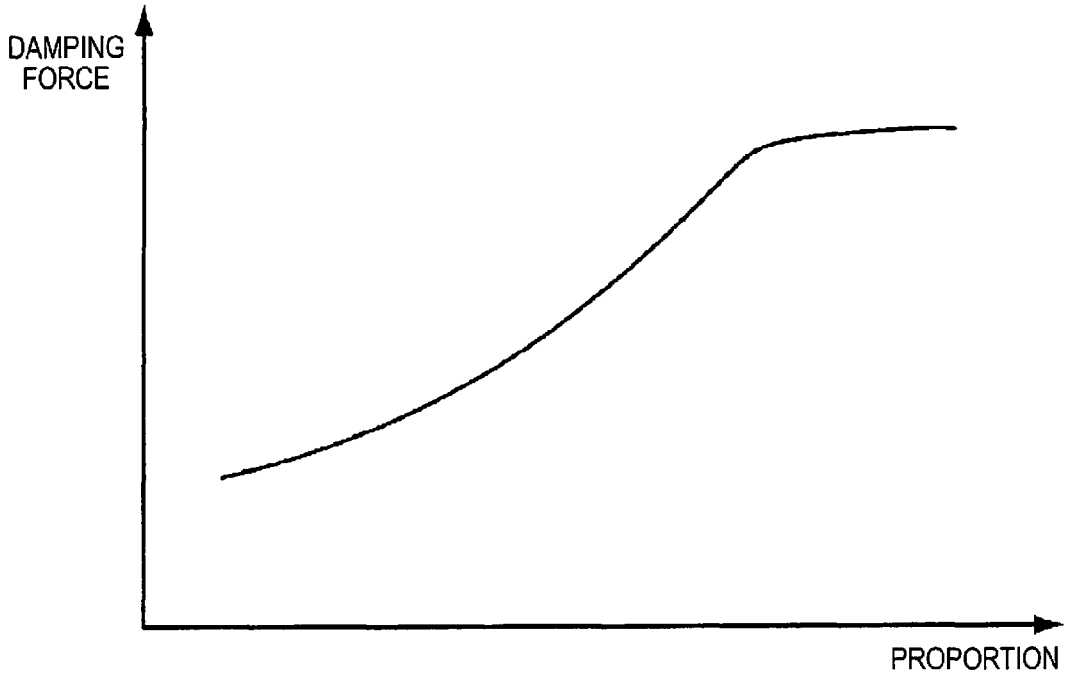
DAMPING FORCE
PROPORTION

FIG. 15

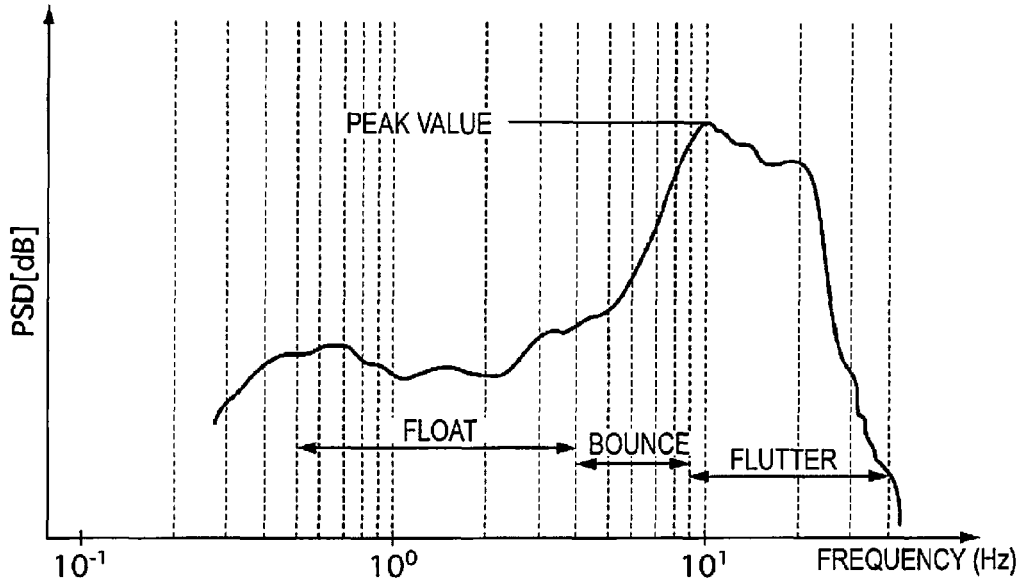

FIG. 16

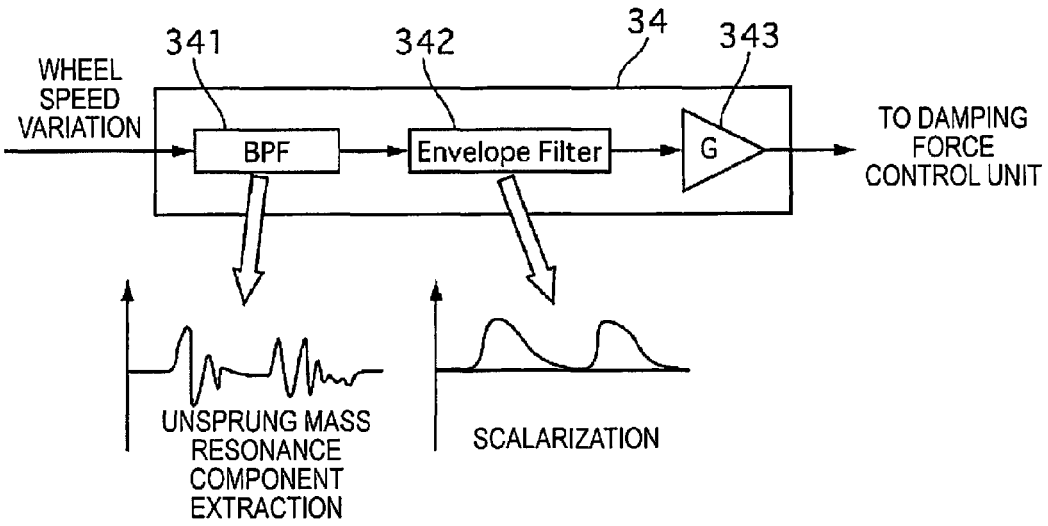

FIG. 17

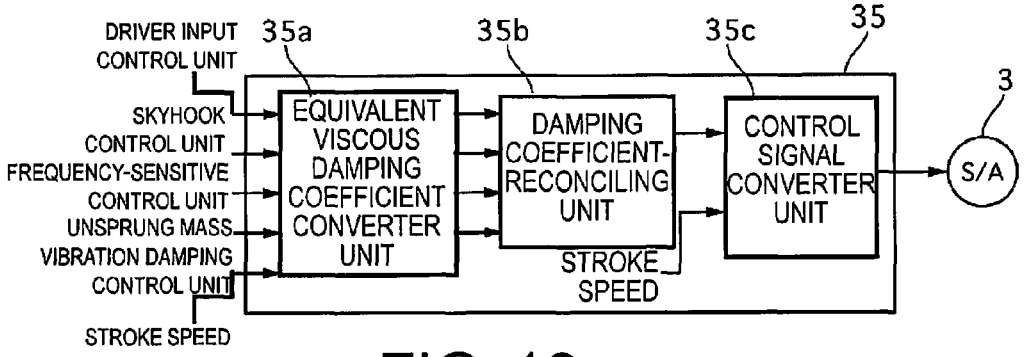

FIG. 18

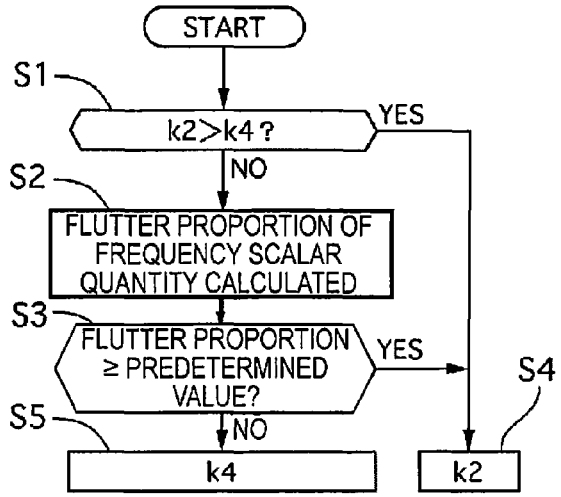

FIG. 19

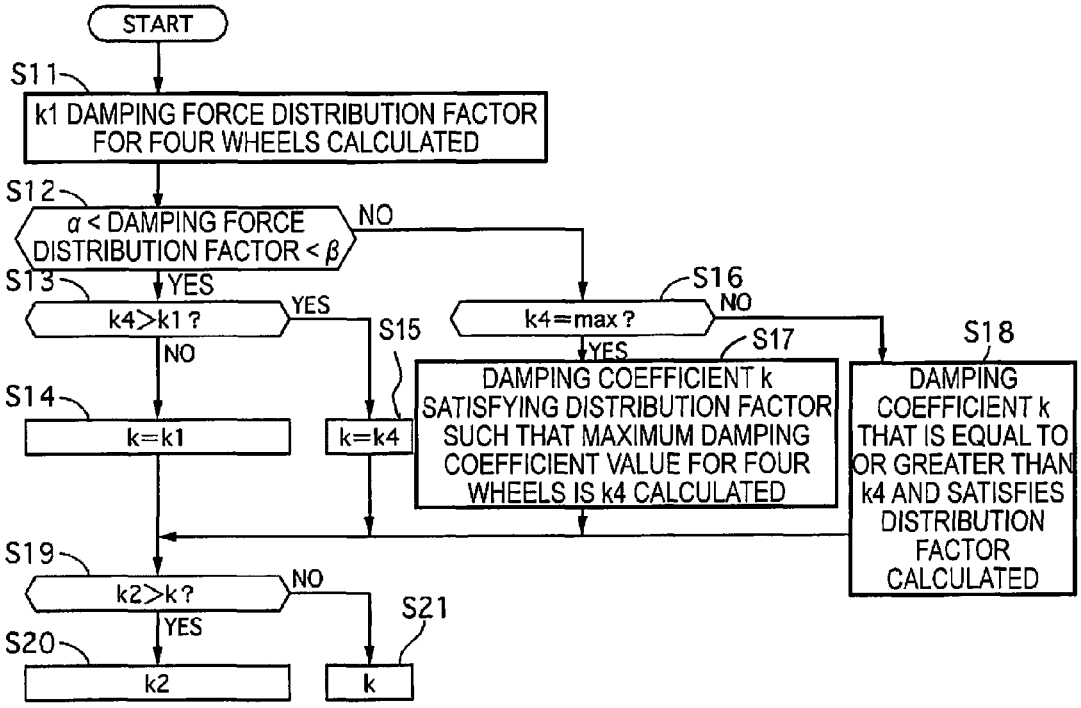

FIG. 20

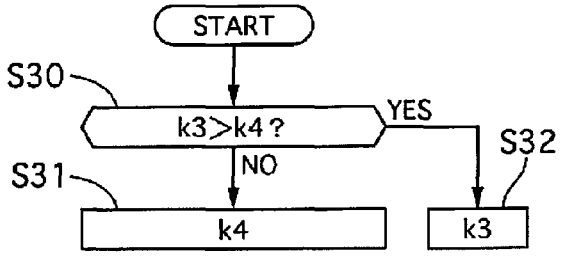

FIG. 21

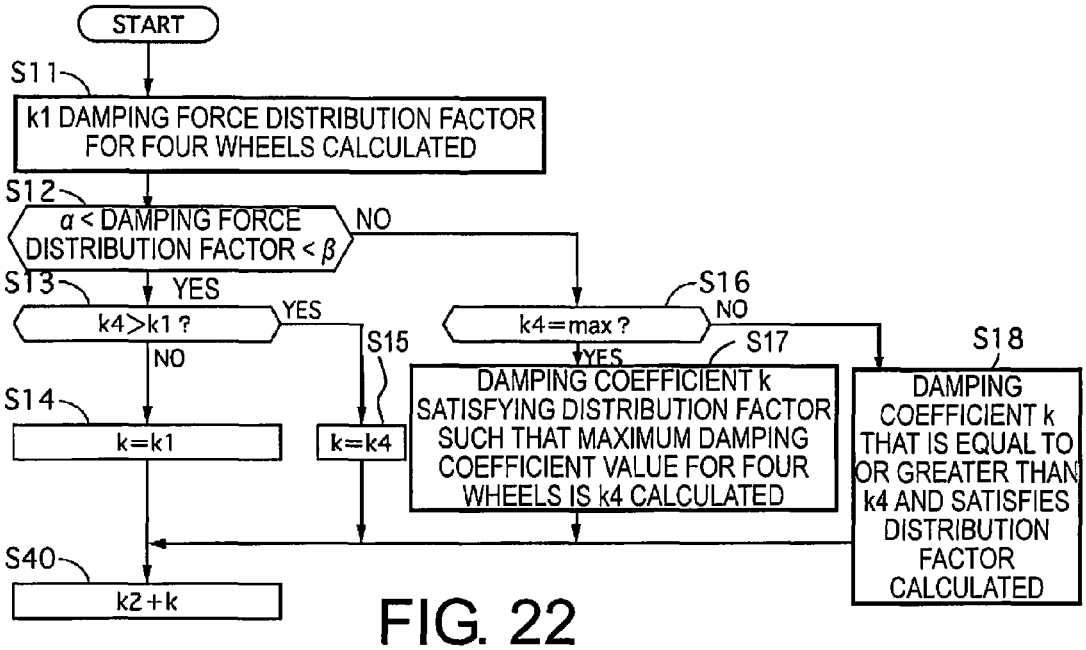

FIG. 22

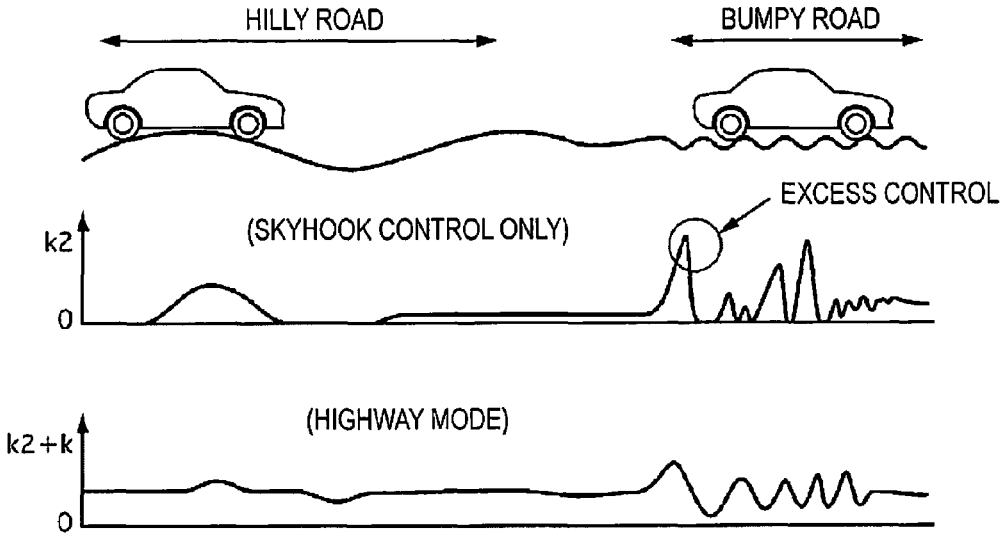

FIG. 23

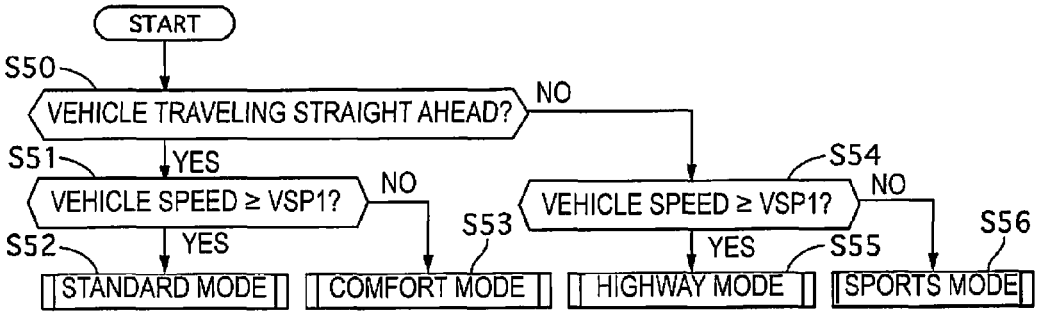

FIG. 24

VEHICLE CONTROL DEVICE

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a U.S. National stage application of International Application No. PCT/JP2013/083828, filed Dec. 27, 2012, which claims priority to Japanese Patent Application No. 2012-013653 filed in Japan on Jan. 26, 2012, the contents of each of which are hereby incorporated herein by reference.

BACKGROUND

1. Field of the Invention

The present invention relates to a control device for controlling the state of a vehicle.

2. Background Information

Japanese Laid-Open Patent Application No. 2009-241813 discloses a technique of estimating the vibrational state of a vehicle being subjected to interference from changes in wheel speed, and altering the damping force of variable-damping-force shock absorbers.

SUMMARY

Because there is little change in wheel speed sensor values in response to axle tilting that occurs as a result of suspension strokes, wheel speed sensors for rear wheels connected to a differential gear exhibit low precision in estimating vehicle vibrational states, leading to reduced vibration damping performance.

The present invention was conceived in view of the problem described above, and has an objective of providing a vehicle control device capable of precisely estimating vehicle vibrational state, and improving vibration damping performance.

In order to achieve the abovementioned objective, a vehicle control device according to the present invention delays a front wheel speed detected by a front wheel speed-detecting device to estimate rear wheel speed, and, taking the sizes of the amplitudes of the frequencies of the detected front wheel speed and rear wheel speed as frequency scalar quantities, calculates the damping force of a variable-damping-force shock absorber on the basis of the frequency scalar quantities.

Wheel speed sensors for front wheels connected to an axle are capable of precisely estimating vehicle vibrational state due to the large changes in wheel speed sensor values that occur in response to axle tilting occurring due to strokes. Because the damping force control amount for the variable-damping-force shock absorbers is calculated on the basis of frequency scalar quantities, there is no reduction in vibration damping performance even if there is a deviation in phase between the estimated rear wheel speed and the actual rear wheel speed, allowing vibration damping performance to be improved using a precisely estimated vehicle vibrational state.

BRIEF DESCRIPTIONS OF THE DRAWINGS

Referring now to the attached drawings which form a part of this original disclosure.

FIG. **1** is a schematic system diagram of a vehicle control device according to a first embodiment.

FIG. **2** is a control block diagram showing a configuration of control performed by the vehicle control device according to the first embodiment.

FIG. **3** is a control block diagram showing a configuration of roll rate minimization control according to the first embodiment.

FIG. **4** is a time chart showing an envelope waveform formation process performed in the roll rate minimization control of the first embodiment.

FIG. **5** is a control block diagram showing the configuration of a driving state estimator unit of the first embodiment.

FIG. **6** is a control block diagram showing the specifics of control in a stroke speed calculator unit of the first embodiment.

FIG. **7** is a block diagram showing the configuration of a reference wheel speed calculator unit of the first embodiment.

FIGS. **8**A and **8**B are flow charts of a rear wheel speed estimation process according to the first embodiment.

FIGS. **9**A and **9**B are schematic diagrams of a vehicle body vibration model.

FIG. **10** is a control block diagram of actuator control amount calculation processes performed during pitch control in the first embodiment.

FIG. **11** is a control block diagram of brake pitch control in the first embodiment.

FIG. **12** is a graph simultaneously showing a wheel speed frequency profile detected by a wheel speed sensor and a stroke frequency profile from a stroke sensor not installed in the present embodiment.

FIG. **13** is a control block diagram showing frequency-sensitive control in sprung mass vibration damping control in the first embodiment.

FIG. **14** is a correlation graph showing human sensation profiles in different frequency regions.

FIG. **15** is a plot showing the relationship between the proportion of vibration contamination and damping force in a float region in the frequency-sensitive control of the first embodiment.

FIG. **16** is a wheel speed frequency profile detected by a wheel speed sensor in certain driving conditions.

FIG. **17** is a block diagram showing a control configuration for unsprung mass vibration damping control in the first embodiment.

FIG. **18** is a control block diagram showing a control configuration for a damping force control unit of the first embodiment.

FIG. **19** is a flow chart of a damping coefficient reconciliation process performed during a standard mode in the first embodiment.

FIG. **20** is a flow chart of a damping coefficient reconciliation process performed during a sports mode in the first embodiment.

FIG. **21** is a flow chart of a damping coefficient reconciliation process performed during a comfort mode in the first embodiment.

FIG. **22** is a flow chart of a damping coefficient reconciliation process performed during a highway mode in the first embodiment.

FIG. **23** is a time chart showing changes in damping coefficient when driving on a hilly road surface and a bumpy road surface.

FIG. **24** is a flow chart of a driving state-based mode selection process performed by a damping coefficient-reconciling unit of the first embodiment.

DETAILED DESCRIPTION OF THE EMBODIMENTS

First Embodiment

FIG. **1** is a schematic system diagram of a vehicle control device according to a first embodiment. A vehicle comprises

an engine **1** constituting a power source, brakes **20** for generating braking torque by applying frictional force to the wheels (brakes corresponding to individual wheels will be referred to hereafter as follows: front right brake: **20**FR; front left brake: **20**FL; rear right brake: **20**RR; rear left brake: **20**RL), and variable-damping-force shock absorbers **3** provided between each of the wheels and the vehicle body ("shock absorber" will be abbreviated "S/A" in the following description; shock absorbers corresponding to individual wheels will be referred to as follows: front right S/A: **3**FR; front left S/A: **3**FL; rear right S/A: **3**RR; rear left S/A: **3**RL).

The engine **1** comprises an engine controller **1***a* (also referred to hereafter as an engine control unit) for controlling the torque outputted by the engine **1**; the engine controller **1***a* controls the engine operation state (engine rpm, engine output torque, etc.) as desired by controlling the opening of the throttle valve, the fuel injection level, the ignition timing, and the like of the engine **1**. The brakes **20** generates braking torque on the basis of hydraulic pressure supplied from a brake control unit **2** capable of controlling brake hydraulic pressure for each of the wheels according to driving state. The brake control unit **2** comprises a brake controller **2***a* (also referred to hereafter as a brake control unit) for controlling the braking torque generated by the brakes **20**; the desired hydraulic pressure is generated in the brakes **20** for each of the wheels by the opening and closing of a plurality of solenoid valves using master cylinder pressure generated by a driver operating the brake pedal or pump pressure generated by a built-in motor-driven pump as a hydraulic pressure source.

The S/As **3** are damping force-generating devices for damping the elastic motion of coil springs provided between the unsprung mass (the axles, wheels, etc.) and the sprung mass (vehicle body, etc.) of the vehicle, and the damping force generated thereby can be adjusted by the operation of actuators. Each of the S/As **3** comprises a cylinder in which fluid is sealed, a piston that makes strokes within the cylinder, and an orifice for controlling the movement of the fluid between fluid chambers formed above and below the piston. Orifices of various diameters are formed in the piston, and an orifice corresponding to a control command is selected from the various orifices when the S/A actuator operates. Damping force corresponding to the diameter of the orifice is thereby generated. The movement of the piston will be more easily restricted if the orifice diameter is small, increasing damping force, and movement of the piston will be less easily restricted if the orifice diameter is large, decreasing damping force.

Apart from selecting the diameter of the orifice, damping force may also be set, for example, by disposing a solenoid control valve over a passage linking the fluid chambers formed above and below the piston and controlling the opening and closing of the solenoid control valve; the invention is not particularly limited with respect thereto. Each of the S/As **3** comprises an S/A controller **3***a* for controlling the damping force of the S/A **3**, and damping force is controlled by the operated of the orifice diameter by the S/A actuator.

Also comprised are wheel speed sensors **5** for detecting the wheel speed of each of the wheels (the sensors will be referred to as follows when wheel speeds corresponding to individual wheels are indicated: front right wheel speed: **5**FR; front left wheel speed **5**FL; rear right wheel speed: **5**RR; rear left wheel speed: **5**RL), an integrated sensor **6** for detecting forward/reverse acceleration, yaw rate, and lateral acceleration acting upon the center of gravity of the vehicle, a steering angle sensor **7** for detecting a steering angle indicating the amount to which the driver has operated the steering wheel, a vehicle speed sensor **8** for detecting vehicle speed, an engine torque sensor **9** for detecting engine torque, an engine rpm sensor **10** for detecting engine rpm, a master pressure sensor **11** for detecting master cylinder pressure, a brake switch **12** for outputting an on state signal when a brake pedal is operated, and an accelerator opening sensor **13** for detecting the degree to which an accelerator pedal is open. Signals from the various sensors are inputted to the S/A controller **3***a*. The integrated sensor **6** may be disposed at the center of gravity of the vehicle or at another location without restriction as long as the sensor is capable of estimating various values at the position of the center of gravity. The sensor need not be integrated; individual sensors for detecting yaw rate, forward/reverse acceleration, and lateral acceleration may also be provided.

Out of the various wheel speed sensors **5**, wheel speed sensors **5**FL, **5**FR connected to the right and left front wheels are connected to a front wheel hub, and axle shaft rotation is detected as wheel speed. Meanwhile, right and left rear wheel speed sensors **5**RL, **5**RR are connected to a differential gear, and rear drive shaft rotation is detected as wheel speed.

FIG. **2** is a control block diagram showing a configuration of control performed by the vehicle control device according to the first embodiment. The first embodiment comprises three controllers: the engine controller **1***a*, the brake controller **2***a*, and the S/A controller **3***a*. The S/A controller **3***a* comprises a driver input control unit **31** for performing driver input control so that a desired vehicle orientation is attained on the basis of driver operations (of the steering wheel, accelerator and brake pedals, etc.), a driving state estimator unit **32** for estimating driving state on the basis of values detected by the sensors, a sprung mass vibration damping control unit **33** for controlling the vibrational state of the sprung mass of the vehicle on the basis of the estimated driving state, an unsprung mass vibration damping control unit **34** for controlling the vibrational state of the unsprung mass of the vehicle on the basis of the estimated driving state, and a damping force control unit **35** for deciding upon the damping force to be set for the S/As **3** on the basis of a shock absorber orientation control amount outputted from the driver input control unit **31**, a sprung mass vibration damping control amount outputted from the sprung mass vibration damping control unit **33**, and an unsprung mass vibration damping control amount outputted from the unsprung mass vibration damping control unit **34**, and controlling S/A damping force.

In the first embodiment, three controllers are provided, but the present invention is not particularly limited to such a configuration; for example, the damping force control unit **35** may be provided separately from the S/A controller **3***a* as an orientation controller for a total of four S/A controllers including the damping force control unit **35**, or all the controllers may be combined into a single integrated controller. The configuration of the first embodiment envisions the repurposing of an engine controller and a brake controller in an existing vehicle as an engine control unit **1***a* and a brake control unit **2***a*, and the installation of a separate S/A controller **3***a* to create the vehicle control device according to the first embodiment.

(Overall Configuration of Vehicle Control Device)

Three actuators are used in the vehicle control device according to the first embodiment in order to control the vibrational state of the sprung mass of the vehicle. Because the control performed by each of the activators affects the state of the sprung mass of the vehicle, interference is a problem. In addition, the elements controllable by the engine **1**, the elements controllable by the brakes **20**, and the elements controllable by the S/As **3** all differ, and the matter of the combinations in which these elements should be controlled is another problem. For example, the brakes **20** are capable of controlling bouncing motion and pitching motion,

but controlling both at the same time will create a strong sense of deceleration and tend to create an unnatural feel for the driver. The S/As **3** are capable of absorbing rolling motion, bouncing motion, and pitching motion, but controlling all three of these using the S/As **3** will lead to increased manufacturing costs for the S/As **3**, and tends to increase damping force; this facilitates high-frequency vibrational input from the road surface, also creating an unnatural feel for the driver. In other words, a trade-off must be made in that control performed by the brakes **20** will not lead to worse high-frequency vibration but will lead to an increased sense of deceleration, and control performed by the S/As **3** will not create a sense of deceleration but will lead to high-frequency vibrational input.

Thus, a control configuration has been adopted for the vehicle control device of the first embodiment in which a comprehensive assessment is made of these problems in order to draw upon the respective advantages of these control methods so as to complement the weaknesses of the other, thereby yielding a vehicle control device that is both economical and offers superior vibration damping performance. To this end, the following points were taken into consideration in the overall construction of the control system.

(1) The amount of control performed by the S/As **3** is minimized by prioritizing control performed by the engine **1** and the brakes **20**.

(2) The only type of motion subjected to control by the brakes **20** is pitching motion, thereby eliminating the sense of deceleration produced from control by the brakes **20**.

(3) The amount of control performed by the engine **1** and the brakes **20** is restricted to less than the actually outputtable control amount, thereby reducing the burden placed upon the S/As **3** and minimizing the unnatural feel yielded by control performed by the engine **1** and the brakes **20**.

(4) Skyhook control is performed by all of the actuators. This allows skyhook control to be inexpensively performed using all of the wheel speed sensors installed in the vehicle, without the use of a stroke sensor, sprung mass vertical acceleration sensor, or the like, as is usually necessary to perform skyhook control.

(5) Scalar control (frequency-sensitive control) has been newly introduced in order to address high-frequency vibrational input, which is difficult to address using skyhook control or other types of vector control, when the S/As **3** are performing sprung mass control.

(6) The control state manifested by the S/As **3** is selected, as appropriate, according to the driving state, thereby providing a control state suited to the driving conditions.

The foregoing has been a summary of the features of the control system according to the embodiment as a whole. The specifics by which each of these individual features will be described in sequence hereafter.

(Driver Input Control Unit)

First, the driver input control unit will be described. The driver input control unit **31** comprises an engine driver input control unit **31***a* for attaining the vehicle orientation demanded by the driver by controlling the torque of the engine **1**, and an S/A driver input control unit **31***b* for attaining the vehicle orientation demanded by the driver by controlling the damping force of the S/As **3**. The engine driver input control unit **31***a* calculates a ground load variation minimization control amount for minimizing variations in the ground loads of the front wheels and the rear wheels, and a yaw response control amount corresponding to the vehicle behavior desired by the driver on the basis of signals from the steering angle sensor **7** and the vehicle speed sensor **8**.

The S/A driver input control unit **31***b* calculates a driver input damping force control amount corresponding to the vehicle behavior desired by the driver on the basis of the signals from the steering angle sensor **7** and the vehicle speed sensor **8**, and outputs this amount to the damping force control unit **35**. If, for example, the nose of the vehicle rises while the driver is turning, the driver's field of view can easily be taken off the road; in such cases, the damping force of the four wheels is outputted as the driver input damping force control amount so as to prevent the nose from rising. A driver input damping force control amount for minimizing rolling generating during turning is also outputted.

(Controlling Rolling Via S/A Driver Input Control)

Roll minimization control performed via S/A driver input control will now be described. FIG. **3** is a control block diagram showing a configuration of roll rate minimization control according to the first embodiment. A lateral acceleration estimator unit **31***b***1** estimates lateral acceleration Yg on the basis of a front wheel steering angle δf detected by the steering angle sensor **7** and a vehicle speed VSP detected by the vehicle speed sensor **8**. The lateral acceleration Yg is calculated according to the following formula on the basis of a vehicle body plan view model:

$$Yg=(VSP^2/(1+A\cdot VSP^2))\cdot\delta f$$

A is a predetermined value.

A 90° phase lead component-generating unit **31***b***2** differentiates the estimated lateral acceleration Yg and outputs a lateral acceleration derivative dYg. A first adding unit **31***b***4** adds the lateral acceleration Yg and the lateral acceleration derivative dYg. A 90° phase lag component-generating unit **31***b***3** outputs a component F(Yg) in which the phase of the lateral acceleration Yg is delayed 90°. A second adding unit **31***b***5** adds F(Yg) to the sum calculated by the first adding unit **31***b***4**. A Hilbert transform unit **31***b***6** calculates a scalar quantity based on the envelope waveform of the obtained sum. A gain multiplier unit **31***b***7** multiplies gain by the envelope waveform-based scalar quantity, calculates a driver input orientation control amount for use in roll rate minimization control, and outputs the amount to the damping force control unit **35**.

FIG. **4** is a time chart showing an envelope waveform formation process performed in the roll rate minimization control of the first embodiment. When a driver begins steering at time t1, a roll rate gradually begins to be generated. At this point, the 90° phase lead component is added to form an envelope waveform, and the driver input orientation control amount is calculated on the basis of the envelope waveform-based scalar quantity, thereby allowing roll rate generation during the initial stage of steering to be minimized. Next, when the driver holds the steering wheel in place at time t2, the 90° phase lead component vanishes, and the phase lag component F(Yg) is then added. At this time, a roll rate resonance component equivalent to the roll aftershock is generated after rolling occurs, even if there is little change in the roll rate itself in a steady steering state. If the phase lag component F(Yg) were not added, damping force would be set to a low value from time t2 to time t3, risking destabilization of vehicle behavior by the roll rate resonance component. The 90° phase lag component F(Yg) is added in order to minimize the roll rate resonance component.

When the driver shifts from the held steering state back to a straight-ahead driving state at time t3, the lateral acceleration Yg decreases, and the roll rate is reduced to a low value. The action of the 90° phase lag component F(Yg) also ensures damping force at this point as well, allowing destabilization due to the roll rate resonance component to be avoided.

(Driving State Estimator Unit)

Next, the driving state estimator unit will be described. FIG. **5** is a control block diagram showing the configuration of a driving state estimator unit of the first embodiment. The driving state estimator unit **32** of the first embodiment calculates a stroke speed, bounce rate, roll rate, and pitch rate for each wheel used in the skyhook control performed by the sprung mass vibration damping control unit **33** as described hereafter primarily on the basis of the wheel speeds detected by the wheel speed sensors **5**. The values from the wheel speed sensors **5** of the wheels are inputted into a stroke speed calculator unit **321**, and sprung mass speed is calculated by the stroke speed calculator unit **321** from the stroke speeds calculated for the wheels.

FIG. **6** is a control block diagram showing the specifics of control in a stroke speed calculator unit of the first embodiment. A stroke speed calculator unit **321** is separately provided for each wheel; the control block diagram shown in FIG. **6** focuses on a specific wheel. The stroke speed calculator unit **321** is provided with: a reference wheel speed calculator unit **300** for calculating two reference wheel speeds (sprung reference wheel speed, unsprung reference wheel speed) on the basis of the values from the wheel speed sensors **5**, a front wheel steering angle δf detected by the steering angle sensor **7**, a rear wheel steering angle δr (the actual rear wheel steering angle if a rear wheel steering device is provided, 0 otherwise), vehicle body lateral speed, and an actual yaw rate detected by the integrated sensor **6**; a tire rotational vibration frequency calculator unit **321***a* for calculating a tire rotational vibration frequency on the basis of the calculated reference wheel speeds; a deviation calculator unit **321***b* for calculating the deviation between the sprung reference wheel speed and the values from the wheel speed sensors (i.e., changes in wheel speed); a GEO conversion unit **321***c* for converting the deviation calculated by the deviation calculator unit **321***b* to a suspension stroke amount; a stroke speed calibrator unit **321***d* for calibrating the converted stroke amount to a stroke speed; a signal processing unit **321***e* for applying a band elimination filter according to the frequency calculated by the tire rotational vibration frequency calculator unit **321***a* to the value calibrated by the stroke speed calibrator unit **321***d* to eliminate a primary tire rotational vibration component, and calculating a final stroke speed; and a deviation calculator unit **321***f* for calculating the deviation between the unsprung reference wheel speed and the values from the wheel speed sensors (i.e., changes in wheel speed).

(Reference Wheel Speed Calculator Unit)

The reference wheel speed calculator unit **300** will now be described.

FIG. **7** is a block diagram of the configuration of a reference wheel speed calculator unit according to a first embodiment. The reference wheel speed calculator unit **300** is provided with a sprung reference wheel speed calculator unit **300***a* for calculating a sprung reference wheel speed used in frequency-sensitive control during sprung mass orientation control, to be described hereafter, and an unsprung reference wheel speed calculator unit **300***b* for calculating an unsprung reference wheel speed used in unsprung vibration damping control, to be described hereafter. The reference wheel speed used here is a wheel speed from which various types of interference have been removed. In other words, the difference between the value from the wheel speed sensor and the reference wheel speed is related to a component that varies according to a stroke generated by vehicle body bouncing motion, rolling motion, pitching motion, or unsprung vertical vibration; in the present embodiment, the stroke speed is calculated on the basis of this difference.

The sprung reference wheel speed calculator unit **300***a* and the unsprung reference wheel speed calculator unit **300***b* are identically configured apart from rear wheel speed estimator units **305***a*, **305***b*, and therefore will be described collectively.

A flat surface motion component extractor unit **301** uses the wheel speed sensor values as inputs to calculate a first wheel speed V0 as a reference wheel speed for each of the wheels on the basis of the vehicle body plan view model. ω (rad/s) is the wheel speed sensor detected by the wheel speed sensor **5**, δf (rad) is a front wheel actual steering angle detected by the steering angle sensor **7**, δr (rad) is a rear wheel actual steering angle, Vx is vehicle body lateral speed, γ (rad/s) is the yaw rate detected by the integrated sensor **6**, V (m/s) is a vehicle body speed estimated from the calculated reference wheel speed ω0, VFL, VFR, VRL, and VRR are the reference wheel speeds to be calculated, Tf is a front wheel treat, Tr is a rear wheel treat, Lf is the distance from the position of the vehicle center of gravity to the front wheels, and Lr is the distance from the position of the vehicle center of gravity to the rear wheel. The vehicle body plan view model is expressed as follows using the symbols described above.

$$VFL=(V-Tf/2\cdot\gamma)\cos\delta f+(Vx+Lf\cdot\gamma)\sin\delta f$$

$$VFR=(V+Tf/2\cdot\gamma)\cos\delta f+(Vx+Lf\cdot\gamma)\sin\delta f$$

$$VRL=(V-Tr/2\cdot\gamma)\cos\delta r+(Vx-Lr\cdot\gamma)\sin\delta r$$

$$VRR=(V+Tr/2\cdot\gamma)\cos\delta r+(Vx-Lr\cdot\gamma)\sin\delta r \quad \text{(Formula 1)}$$

If a normal driving state in which no lateral sliding of the vehicle occurs is hypothesized, 0 may be inputted for the vehicle body lateral speed Vx. This yields the following formulas when the various formulas are rewritten with values based on V. When rewriting in this manner, V is written as V0FL, V0FR, V0RL, and V0RR (equivalent to first wheel speeds) as values corresponding to the various wheels.

$$V0FL=\{VFL-Lf\cdot\gamma\sin\delta f\}/\cos\delta f+Tf/2\cdot\gamma$$

$$V0FR=\{VFR-Lf\cdot\gamma\sin\delta f\}/\cos\delta f-Tf/2\cdot\gamma$$

$$V0RL=\{VRL+Lr\cdot\gamma\sin\delta r\}/\cos\delta r+Tr/2\cdot\gamma$$

$$V0RR=\{VRR+Lf\cdot\gamma\sin\delta f\}/\cos\delta R-Tr/2\cdot\gamma \quad \text{(Formula 2)}$$

A sprung rear wheel speed estimator unit **305***a* and an unsprung rear wheel speed estimator unit **305***b* delay the first wheel speed for the front wheels to calculated an estimated first wheel speed for the rear wheels, select either the estimated first wheel speed or the rear wheel first wheel speed outputted by the flat surface motion component extractor unit **301** according to vehicle speed, and outputs the selected speed as the first wheel speed for the rear wheels. The estimated first wheel speed for the left rear wheel is calculated from the first wheel speed for the left front wheel, and the estimated first wheel speed for the right rear wheel is calculated from the first wheel speed for the right front wheel. If predetermined conditions are met, the sprung rear wheel speed estimator unit **305***a* and the unsprung rear wheel speed estimator unit **305***b* store the first wheel speed for the front wheels in memory, and output the front wheel first wheel speed stored in memory instead of the rear wheel first wheel speed outputted by the flat surface motion component extractor unit **301** after a delay time has elapsed as the first wheel speed for the rear wheels. On the other hand, if the predetermined conditions are not met, the front wheel first wheel speed is not stored. The "delay time" referred to here is the

wheel base divided by vehicle speed. When turning (yaw rate≠0), the yaw rate is taken into account in calculating the delay time.

The following conditions 1 and 2 are the predetermined conditions for the sprung rear wheel speed estimator unit **305***a*, and the following condition 1 for the unsprung rear wheel speed estimator unit **305***b*.

1. Vehicle speed is equal to or greater than a low vehicle speed threshold value V1 and lower than a high vehicle speed threshold value V2

2. Frequency-sensitive control performed as sprung mass orientation control

The low vehicle speed threshold value V1 is a vehicle speed slightly higher than a vehicle speed at which the amount of stored information exceeds the upper limit of a pre-reserved buffer region when the vehicle is driving at a constant vehicle speed. The high vehicle speed threshold value V2 is a vehicle speed slightly lower than a speed obtained by dividing wheel base by the sampling cycle for the wheel speed sensor values.

In this embodiment, the sprung rear wheel speed estimator unit **305***a* and the unsprung rear wheel speed estimator unit **305***b* are disposed between the flat surface motion component extractor unit **301** and a road surface interference-eliminating unit, but they may also be disposed before the flat surface motion component extractor unit **301**.

FIGS. **8**A and **8**B are flow charts showing a rear wheel speed estimation process performed by the sprung rear wheel speed estimator unit **305***a*, with (a) being a process of storing a first wheel speed for the front wheels in memory and (b) being a process of outputting a first wheel speed for the rear wheels. The two processes are repeatedly executed independently in each sampling cycle.

In step S**61**, a first wheel speed for the front wheels is inputted.

In step S**62**, it is determined whether the vehicle speed at the time the front wheel first wheel speed was inputted in step S**61** is equal to or greater than the low vehicle speed threshold value V1 and less than the high vehicle speed threshold value V2; if so, the process continues to step S**63**, and if not, the process continues to RETURN.

In step S**63**, it is determined whether frequency-sensitive control is being performed as sprung vibration damping control; if so, the process continues to step S**64**, and if not, the process continues to RETURN.

In step S**64**, a delay time (i.e., how many cycles to wait before outputting the front wheel first wheel speed inputted in step S**61**) is calculated by dividing the wheel base by vehicle speed. If the vehicle is turning at this time, the yaw rate is taken into account.

In step S**65**, the front wheel first wheel speed inputted in step S**61** and the delay time are stored in the memory as delay information.

In step S**66**, it is determined whether the front wheel first wheel speed outputted in said sampling cycle is stored in the memory; if so, the process continues to step S**67**, and if not, the process continues to step S**68**.

In step S**68**, the front wheel first wheel speed stored in the memory is outputted as a rear wheel first wheel speed.

In step S**69**, the rear wheel first wheel speed inputted by the flat surface motion component extractor unit **301** is outputted.

In the rear wheel speed estimation process performed by the sprung rear wheel speed estimator unit **305***a* as described above, rear wheel estimation is prohibited if the vehicle speed is less than the low vehicle speed threshold value V1 because a memory buffer overload will occur. If the vehicle speed is equal to or greater than the high vehicle speed threshold value V2, rear wheel estimation is prohibited because an impossible delay time will be set. In addition, rear wheel estimation is prohibited when the vehicle is in reverse because it is impossible to estimate rear wheel speed using values from the front wheel speed sensors. Rear wheel estimation is also prohibited if skyhook control is being performed as sprung vibration damping control, as vibration damping performance may be negatively affected.

The rear wheel speed estimation process performed by the unsprung rear wheel speed estimator unit **305***b* is identical, except that step S**63** is eliminated from FIG. **8**A; thus, description thereof will be omitted. Because the unsprung rear wheel speed is not used in sprung vibration damping control, it has no effect whatsoever even if skyhook control is being performed.

A roll interference-removing unit **302** uses the first wheel speed V0 as an input to calculate second wheel speeds V0F, V0R as reference wheel speeds for the front and rear wheels on the basis of a vehicle body front view model. The vehicle body front view model is used to remove wheel speed differences generated by rolling motion occurring around a center of roll rotation on a normal line passing through the vehicle center of gravity as viewed from the front of the vehicle, and is represented as follows.

$$V0F=2(V0\mathrm{FL}+V0\mathrm{FR}2)/2$$

$$V0R=2(V0\mathrm{RL}+V0\mathrm{RR}2)/2$$

This yields second wheel speeds V0F, V0R from which roll-based interference has been removed.

A pitch-interference-removing unit **303** uses the second wheel speeds V0F, V0R as inputs to calculate third wheel speeds VbFL, VbFR, VbRL, and VbRR constituting reference wheel speeds for all the wheels according to a vehicle body side view model. The vehicle body side view model is used to remove wheel speed differences generated by pitching motion occurring around a center of pitch rotation on a normal line passing through the vehicle center of gravity as viewed from the side of the vehicle.

$$V\mathrm{bFL}=V\mathrm{bFR}=V\mathrm{bRL}=V\mathrm{bRR}=\{Lr/(Lf+Lr)\}V0F+\{Lf/(Lf+Lr)\}V0R \quad \text{(Formula 3)}$$

A reference wheel speed redistribution unit **304** uses VbFL (=VbFR=VbRL=VbRR) for V in the vehicle body plan view model shown in formula 1 to calculate final reference wheel speeds VFL, VFR, VRL, VRR for each wheel, which are divided by the tire radius r0 to calculate the sprung mass reference wheel speed ω0.

Once the reference wheel speed ω0 has been calculated according to the process described above, the deviation between the reference wheel speed ω0 and the wheel speed sensor value is calculated; the deviation represents a wheel speed variation arising from suspension strokes, and is therefor converted into a stroke speed Vz_s. As a rule, not only does a suspension make strokes in the vertical direction when holding the wheels, but the wheel rotational centers also move forwards and backwards as strokes occur, and the axles become tilted, creating a difference in rotational angle with the wheels. Because this forward and backward motion leads to changes in wheel speed, deviations between the reference wheel speed and the wheel speed sensor value can be extracted as stroke-induced variations. The degree of variation that occurs can be set, as appropriate, according to the suspension geometry.

Because the front wheel speed sensors **5**FL, **5**FR are connected to the axle (front wheel hub), there is a large change in wheel speed sensor values in response to suspension stroke-induced axle tilting. By contrast, because the rear wheel

speed sensors **5**RL, **5**RR are connected to a differential gear, there is little change in wheel speed sensor values in response to suspension stroke-induced axle tilting, and such axle tilting is not readily reflected in wheel speed sensor values. In other words, the rear wheel speed sensors **5**RL, **5**RR exhibit little sensitivity to suspension stroke-induced axle tilting. Therefore, using wheel speed sensor values from the rear wheel speed sensors **5**RL, **5**RR to calculate the reference wheel speed will reduce stroke speed estimation precision, negatively affecting vibration damping performance.

Thus, in the first embodiment, the wheel speed detected by the front wheel speed sensors **5**FL, **5**FR, which are highly sensitive to suspension stroke-induced axle tilting, is delayed to estimate the rear wheel speed, thereby increasing stroke speed estimation precision, and minimizing reductions in vibration damping performance.

Once the stroke speed calculator unit **321** has calculated the stroke speeds Vz_sFL, Vz_sFR, Vz_sRL, Vz_sRR for each wheel according to the process described above, a sprung mass speed calculator unit **322** calculates a bounce rate, roll rate, and pitch rate for use in skyhook control.

(Estimation Model)

In skyhook control, damping force is set according to the relationship between the stroke speeds of the S/As **3** and the sprung mass speed, and the orientation of the sprung mass is controlled to achieve a flat driving state. In order to achieve control of the orientation of the sprung mass via skyhook control, feedback on the sprung mass speed is necessary. Stroke speed is a value detectable from the wheel speed sensor **5**; since a sensor for the vertical acceleration of the sprung mass is not provided, the sprung mass speed must be estimated using an estimation model. Problems involved in the estimation model and the appropriate model configuration to adopt will now be discussed.

FIGS. **9**A and **9**B are schematic diagrams of a vehicle body vibration model. FIG. **9**A is a model for a vehicle provided with S/As of constant damping force (hereafter referred to as a conventional vehicle), and FIG. **9**B is a model for a vehicle provided with variable S/As in which skyhook control is performed. In FIGS. **9**A and **9**B, Ms indicates sprung mass, Mu indicates unsprung mass, Ks indicates coil spring modulus of elasticity, Cs indicates S/A damping coefficient, Ku indicates unsprung (tire) modulus of elasticity, Cu indicates unsprung (tire) damping coefficient, and Cv indicates a variable damping coefficient. z2 indicates the position of the sprung mass, z1 indicates the position of the unsprung mass, and z0 indicates the position of the road surface.

If the conventional vehicle model shown in FIG. **9**A is used, the equation of motion for the sprung mass is expressed as follows. The first-order differential for z1 (i.e., speed) is represented by dz1, and the second-order differential (i.e., acceleration) is represented by ddz1.

$$Ms \cdot ddz2 = -Ks(z2-z1) - Cs(dz2-dz1) \quad \text{(Estimation formula 1)}$$

Applying a Laplace transform to this relationship yields the following formula.

$$dz2 = -(1/Ms)\cdot(1/S^2)\cdot(Cs\cdot s+Ks)(dz2-dz1) \quad \text{(Estimation formula 2)}$$

Because dz2−dz1 is stroke speed (Vz_sFL, Vz_sFR, Vz_sRL, Vz_sRR), the sprung mass speed can be calculated from the stroke speed. However, adjusting damping force via skyhook control will vastly reduce estimation precision, creating the problem that a large orientation control force (damping force adjustment) cannot be applied in the conventional vehicle model.

Thus, the use of a skyhook control-based vehicle model shown in FIG. **9**B is conceivable. As a rule, altering damping force involves altering the force limiting the piston movement speed of the S/As **3** as suspension strokes occur. Because semi-active S/As **3** in which the pistons cannot be actively moved in a desired direction, a semi-active skyhook model is used; calculating sprung mass speed yields the following formula.

$$dz2 = -(1/Ms)\cdot(1/s^2)\cdot\{(CS+CV)\cdot s+Ks\}(dz2-dz1) \quad \text{(Estimation formula 3)}$$

wherein:

if $dz2\cdot(dz2-dz1)\geq 0, Cv = Csky\cdot\{dz21(dz2-dz1)\}$, and

if $dz2\cdot(dz2-dz1)<0, Cv=0$.

That is, Cv is a discontinuous value.

If the semi-active skyhook model is viewed as a filter when one wishes to estimate sprung mass speed using a simple filter, the variables are equivalent to filter coefficients, and a variable damping coefficient Cv that is discontinuous in the pseudo-differential term {(Cs+Cv)·s+Ks}; thus, filter response is unstable, and suitable estimation precision cannot be obtained. In particular, unstable filter response will lead to phase shifting. Skyhook control cannot be achieved if the correspondence between the phase and the sign for sprung mass speed breaks down. The decision was thus made to estimate sprung mass speed using an active skyhook model, in which a stable Csky can be directly used without relying upon the sign relationship between sprung mass speed and the stroke speed even if semi-active S/As **3** are used. The use of an active skyhook model to calculate sprung mass speed can be expressed as follows.

$$dz2 = -(1/s)\cdot\{1/(s+Csky/Ms)\}\cdot\{(Cs/Ms)s+(Ks/Ms)\}(dz2-dz1) \quad \text{(Estimation formula 4)}$$

In this case, there is no discontinuity in the pseudo-differential term {(Cs/Ms)s+(Ks/Ms)}, and the term {1/(s+Csky/Ms)} can be constituted using a low-pass filter. Filter response is therefore stable, and suitable estimation precision is obtainable. It should be noted that, even if an active skyhook model is adopted, only semi-active control is actually possible; thus, the controllable range is halved. The estimated sprung mass speed is therefore less than the actual speed in the frequency band from sprung mass resonance down. However, it is the phase that is most vital in the context of skyhook control, and skyhook control can be achieved as long as the correspondence between phase and sign can be maintained; this is unproblematic because the sprung mass speed can be adjusted using the other coefficients or the like.

It is apparent from the relationship described above that sprung mass speed can be estimated if the stroke speeds for each wheel are known. Because an actual vehicle has not one wheel, but four, we will now consider using the stroke speeds for each wheel to estimate the state of the sprung mass divided into roll rate, pitch rate, and bounce rate modes. When calculating the abovementioned three components from the stroke speeds of the four wheels, one corresponding component is lacking, leading to an indefinite solution; thus, warp rate, which indicates the movement of diagonally opposed wheels, has been introduced. Defining xsB as the bounce term of the stroke amount, xsR as the roll term, xsP as the pitch term, xsW as the warp term, and z_sFL, z_sFR, z_sRL, z_sRR as stroke amounts corresponding to Vz_sFL, Vz_sFR, Vz_sRL, Vz_sRR, the following formula arises.

$$\begin{Bmatrix} z_sFL \\ z_sFR \\ z_sRL \\ z_sRR \end{Bmatrix} = \begin{bmatrix} 1 & 1 & -1 & -1 \\ 1 & -1 & -1 & 1 \\ 1 & 1 & 1 & 1 \\ 1 & -1 & 1 & -1 \end{bmatrix} \begin{Bmatrix} xsB \\ xsR \\ xsP \\ xsW \end{Bmatrix} \Rightarrow \begin{Bmatrix} xsB \\ xsR \\ xsP \\ xsW \end{Bmatrix} = \begin{bmatrix} 1 & 1 & -1 & -1 \\ 1 & -1 & -1 & 1 \\ 1 & 1 & 1 & 1 \\ 1 & -1 & 1 & -1 \end{bmatrix}^{-1} \begin{Bmatrix} z_sFL \\ z_sFR \\ z_sRL \\ z_sRR \end{Bmatrix} \quad \text{(Formula 1)}$$

In view of the relationship shown above, the differentials dxsB, . . . of xsB, xsR, xsP, xsW are expressed by the following formulas.

$$dxsB = \tfrac{1}{4}(Vz_sFL + Vz_sFR + Vz_sRL + Vz_sRR)$$

$$dxsR = \tfrac{1}{4}(Vz_sFL - Vz_sFR + Vz_sRL - Vz_sRR)$$

$$dxsP = \tfrac{1}{4}(-Vz_sFL - Vz_sFR + Vz_sRL + Vz_sRR)$$

$$dxsW = \tfrac{1}{4}(-Vz_sFL + Vz_sFR + Vz_sRL - Vz_sRR)$$

The relationship between sprung mass speed and stroke speed is obtained from estimation formula 4 above; thus, taking G as the section of estimation formula 4 reading $-(1/s)\cdot\{1/(s+Csky/Ms)\}\cdot\{(Cs/Ms)s+(Ks/Ms)\}$, GB, GR, and GP as values taking into account modal parameters (CskyB, CskyR, CskyP, CsB, CsR, CsP, KsB, KsR, KsP) for the bounce terms, roll terms, and pitch terms of Csky, Cs, and Ks, respectively, dB as bounce rate, dR as roll rate, and dP as pitch rate, dB, dR, and dP can be calculated as follows.

$$dB = GB \cdot dxsB$$

$$dR = GR \cdot dxsR$$

$$dP = GP \cdot dxsP$$

As shown from the foregoing, the state of the sprung mass of an actual vehicle can be estimated on the basis of the stroke speeds for the various wheels.

(Sprung Mass Vibration Damping Control Unit)

Next, the configuration of the sprung mass vibration damping control unit **33** will be described. As shown in FIG. **2**, the sprung mass vibration damping control unit **33** comprises a skyhook control unit **33***a* for controlling orientation according to the estimated value for sprung mass speed described above, and a frequency-sensitive control unit **33***b* for minimizing sprung mass vibration on the basis of the road surface input frequency.

(Configuration of Skyhook Control Unit)

The vehicle control device according to the first embodiment comprises three actuators for achieving sprung mass orientation control in the form of the engine **1**, the brakes **20**, and the S/As **3**. Of these, bounce rate, roll rate, and pitch rate are the objects of control for the S/As **3**, bounce rate and pitch rate are the objects of control for the engine **1**, and pitch rate is the object of control for the brakes **20** in the skyhook control unit **33***a*. In order to allocate control amounts to a plurality of actuators that act in different manners and control the state of the sprung mass, a shared control amount must be used for each. In the first embodiment, the control amount for each of the actuators can be determined by using the sprung mass speed estimated by the driving state estimator unit **32**.

Bounce-directional skyhook control amount:

$$FB = CskyB \cdot dB$$

Roll-directional skyhook control amount:

$$FR = CskyR \cdot dR$$

Pitch-directional skyhook control amount:

$$FP = CskyP \cdot dP$$

FB is sent to the engine **1** and the S/As **3** as the bounce orientation control amount. FR is for control performed only by the S/As **3**, and so is sent to the damping force control unit **35** as a roll orientation control amount.

Next, the pitch-directional skyhook control amount FP will be described. Pitch control is performed by the engine **1**, the brakes **20**, and the S/As **3**.

FIG. **10** is a control block diagram of actuator control amount calculation processes performed during pitch control in the first embodiment. The skyhook control unit **33***a* comprises a first target orientation control amount calculator unit **331** for calculating a target pitch rate constituting a first target orientation control amount that is a control amount that can used in common for all of the actuators, an engine orientation control amount calculator unit **332** for calculating the engine orientation control amount achieved by the engine **1**, a brake orientation control amount calculator unit **334** for calculating the brake orientation control amount achieved by the brakes **20**, and an S/A orientation control amount calculator unit **336** for calculating the S/A orientation control amount achieved by the S/As **3**.

As the skyhook control according to the present system gives foremost priority to pitch rate minimization, the first target orientation control amount calculator unit **331** outputs pitch rate without further modification (hereafter, this pitch rate will be referred to as the first target orientation control amount). The engine orientation control amount calculator unit **332** calculates the engine orientation control amount, which is the control amount achievable by the engine **1**, on the basis of the inputted first target orientation control amount.

A limit value that limits the engine torque control amount according to the engine orientation control amount so as not to create an unnatural sensation for the driver is set in the engine orientation control amount calculator unit **332**. The engine torque control amount is thus kept within a predetermined forward/reverse acceleration range when converted to forward/reverse acceleration. Accordingly, when the engine torque control amount is calculated on the basis of the first target orientation control amount and results in a value that is equal to or greater than the limit value, a skyhook control amount for the pitch rate attainable using the limit value (i.e., the product of the pitch rate minimized by the engine **1** multiplied by CskyP; hereafter, "engine orientation control amount") is outputted. At this time, a value converted to the pitch rate by a converter unit **332***a* is outputted to a second target orientation control amount calculator unit **333**, to be described hereafter. In addition, the engine control unit **1***a* calculates the engine torque control amount on the basis of the engine orientation control amount corresponding to the limit value, and outputs this amount to the engine **1**.

The second target orientation control amount calculator unit **333** calculates the difference between the first target orientation control amount and the value of the engine orientation control amount converted to pitch rate by the converter unit **332***a*, and outputs this difference to the brake orientation control amount calculator unit **334**. A limit value for limiting the braking torque control amount so as not to create an unnatural feel for the driver, as in the case of the engine **334**, is set in the brake orientation control amount calculator unit **334** (this limit value will be discussed in detail hereafter).

The braking torque control amount is thus kept within a predetermined forward/reverse acceleration range when converted to forward/reverse acceleration (with the limit value being calculated on the basis of naturalness of ride feel for passengers, actuator lifespan, etc.). Accordingly, when the brake orientation control amount is calculated based on the second target orientation control amount, and the calculated value is equal to or greater than the limit value, a pitch rate minimization amount (hereafter, "brake orientation control amount") achievable by such a limit value is outputted. At this time, a value converted to the pitch rate by a converter unit **3344** is outputted to a third target orientation control amount calculator unit **335**, to be described hereafter. In addition, the brake control unit **2***a* calculates a braking torque control amount (or deceleration) on the basis of the brake orientation control amount corresponding to the limit value, and is outputted to the brake control unit **2**.

The third target orientation control amount calculator unit **335** calculates a third target orientation control amount constituted by the difference between the second target orientation control amount and the brake orientation control amount, and outputs this amount to the S/A orientation control amount calculator unit **336**. The S/A orientation control amount calculator unit **336** outputs a pitch orientation control amount corresponding to the third target orientation control amount.

The damping force control unit **35** calculates the damping force control amount on the basis of the bounce orientation control amount, the roll orientation control amount, and the pitch orientation control amount (hereafter collectively referred to as the S/A orientation control amounts), and outputs this amount to the S/As **3**.

(Brake Pitch Control)

Brake pitch control will now be described. Generally, the brakes **20** are capable of controlling both bounce and pitch; thus, it is preferable that they control both. However, when bounce control is performed by the brakes **20**, braking force is applied to all four wheels simultaneously, and there is a strong sense of deceleration even in directions of low control priority despite the difficulty in obtaining control effects, tending to create an unnatural feed for the driver. Thus, a configuration in which the brakes **20** specialize in pitch control has been adopted. FIG. **10** is a control block diagram of brake pitch control in the first embodiment. Defining m as the mass of the vehicle body, Bff as front wheel braking force, BFr as rear wheel braking force, Hcg as the height between the vehicle center of gravity and the road surface, a as vehicle acceleration, Mp as pitch moment, and Vp as pitch rate, the following relationships hold.

$$BFf+BFr=m\cdot a$$

$$m\cdot a\cdot Hcg=Mp$$

$$Mp=(BFf+BFr)\cdot Hcg$$

If braking force is applied when the pitch rate Vp is positive, i.e., the front wheel side of the vehicle is lowered, the front wheel side will sink further lower, augmenting pitch motion; thus, braking force is not applied in such cases. On the other hand, when the pitch rate Vp is negative, i.e., the front wheel side of the vehicle is raised, the braking pitch moment will impart braking force, minimizing the rising of the front wheel side. This ensures the driver's field of view and makes the area ahead easier to see, contributing to improved senses of safety and flatness of ride. In other words, the following control amounts are applied:

when $Vp>0$ (front wheels lowered), $Mp=0$; and

when $Vp\leq 0$ (front wheels raised), $Mp=CskyP\cdot Vp$

Braking torque is thus generated only when the front side of the vehicle is raised, thereby allowing the sense of deceleration created thereby to be reduced compared to cases in which braking torque is generated both when the front side is raised and when it is lowered. In addition, the actuators need only be operated at half the frequency as usual, allowing inexpensive actuators to be used.

The brake orientation control amount calculator unit **334** comprises the following control blocks on the basis of the relationship described above. A dead band process sign determiner unit **3341** determines the sign for the inputted pitch rate Vp; if the sign is positive, no control is necessary, so 0 is outputted to a deceleration sense-reducing processor **3342**, and if the sign is negative, control is determined to be possible, so a pitch rate signal is outputted to the deceleration sense-reducing processor **3342**.

(Deceleration Sense Reduction Process)

Next, a deceleration sense reduction process will be described. This process corresponds to the limit applied by the limit value set in the brake orientation control amount calculator unit **334**. A squaring processor **3342***a* squares the pitch rate signal. This reverses the sign, and smoothes the increase in control force. A pitch rate square damping moment calculator unit **3342***b* multiplies the squared pitch rate by a skyhook gain CskyP for the pitch term that takes the squaring process into account to calculate pitch moment Mp. A target deceleration calculator unit **3342***c* divides the pitch moment Mp by mass m and the Height Hcg between the vehicle center of gravity and the road surface to calculate a target deceleration.

A jerk threshold value limiter unit **3342***d* determines whether the rate of change in the calculated target deceleration, i.e., jerk, is within preset deceleration jerk threshold value and release jerk threshold value ranges, and whether the target deceleration is within a forward/reverse acceleration limit value range. If any of the threshold values is exceeds, the target deceleration is corrected to a value within the ranges for the jerk threshold values. If the target deceleration exceeds the limit value, it is set to within the limit value. It is thereby possible to generate deceleration so as not to create an unnatural feel for the driver.

A target pitch moment converter unit **3343** multiplies the target deceleration limited by the jerk threshold value limiter unit **3342***d* by the mass m and the height Hc9 to calculate a target pitch moment, which is outputted to the brake control unit **2***a* and a target pitch rate converter unit **3344**. The target pitch rate converter unit **3344** divides the target pitch moment by the pitch term skyhook gain CskyP to convert to a target pitch rate (equivalent to a brake orientation control amount), which is outputted to a third target orientation control amount calculator unit **335**.

For pitch rate, as described above, the first target orientation control amount is calculated, the engine orientation control amount is next calculated, the brake orientation control amount is calculated from the second target orientation control amount constituted by the difference between the first target orientation control amount and the engine orientation control amount, and the S/A orientation control amount is calculated from the third target orientation control amount constituted by the difference between the second orientation control amount and the brake orientation control amount. It is thereby possible to reduce the amount of pitch rate control performed by the S/As **3** by the control performed by the engine **1** and the brakes **20**, thus allowing for a relative reduction in the controllable range of the S/As **3**, and sprung mass orientation control to be achieved using inexpensive S/As **3**.

An increase in the amount of control performed by the S/As **3** will, in general, increase damping force. Because increased damping force leads to stiff suspension characteristics, high-frequency input is more easily transmitted when there is high-frequency vibration input from the road surface, reducing passenger comfort (this situation is hereafter referred to as exacerbated high-frequency vibration characteristics). By contrast, controlling pitch rate using the engine **1** and the brakes **20**, which are actuators whose vibration transmission profiles are not affected by road surface inputs, and reducing the amount of control performed by the S/As **3** allows the exacerbation of high-frequency vibration characteristics to be avoided. The effects described above are obtained by determining the amount of control performed by the engine **1** before that performed by the S/As **3**, and the amount of control performed by the brakes **20** before that performed by the S/As **3**.

(Frequency-Sensitive Control Unit)

Next, a frequency-sensitive control process performed in the sprung mass vibration damping control unit will be described. In the first embodiment, as a rule, the sprung mass speed is estimated on the basis of the values detected by the wheel speed sensors **5**, and skyhook control is performed based thereon, thereby achieving sprung mass vibration damping control. However, there are cases in which it may not be possible to guarantee sufficient estimation precision using the wheel speed sensors **5**, and cases in which it is desirable to actively guarantee a comfortable driving state (i.e., a soft ride rather than a feeling of vehicle body flatness) depending on driving conditions or the driver's intent. In such cases, it may be difficult to effect suitable control due to slight phase shifts if vector control such as skyhook control, in which the relationship (phase, etc.) of the signs of the stroke speed and the sprung mass speed is vital, is used; thus, frequency-sensitive control constituted by sprung mass vibration damping control according to vibration profile scalar quantities has been introduced.

FIG. **12** is a graph simultaneously showing a wheel speed frequency profile detected by a wheel speed sensor and a stroke frequency profile from a stroke sensor not installed in the present embodiment. In this context, "frequency profile" refers to a profile in which the magnitude of amplitude against frequency is plotted on the y axis. A comparison of the frequency component of the wheel speed sensor **5** and the frequency component of the stroke sensor shows that roughly similar scalar quantities can be plotted from the sprung mass resonance frequency component to the unsprung mass resonance frequency component. Thus, the damping force has been set on the basis of this frequency profile out of the values detected by the wheel speed sensor **5**. The region in which the sprung mass resonance frequency component is present is a frequency region in which the swaying of a passenger's entire body creates a sensation as thought the passenger is floating in the air, that is, that the gravitational acceleration affecting the passenger has decreased, and is referred to as the float region (0.5-3 Hz). The region between the sprung mass resonance frequency component and the unsprung mass resonance frequency component is a frequency region in which, although there is no sensation of reduced gravitational acceleration, there is a sensation similar to the quick, frequent bouncing experienced by a person on horseback when riding at a trot, that is, an up-and-down motion followed by the entire body, and is referred to as the bounce region (3-6 Hz). The region in which the unsprung mass resonance frequency component is present is a frequency region in which, although vertical body mass movement is not experienced, frequent vibrations are conveyed to a part a passenger's body, such as the thighs, and is referred to as a flutter region (6-23 Hz).

FIG. **13** is a control block diagram showing frequency-sensitive control in sprung mass vibration damping control in the first embodiment. A band elimination filter **350** cuts out noise apart from the frequency component in the wheel speed sensor value that is used to perform control. A predetermined frequency region splitter unit **351** splits the frequency component into a float region, a bounce region, and a flutter region. A Hilbert transform processor unit **352** performs a Hilbert transform upon the split frequency bands, converting them to scalar quantities (specifically, areas calculated using amplitude and frequency band) based on the amplitude of the frequency.

A vehicle vibrational system weight-setting unit **353** sets weights for the vibration from the float region, bounce region, and flutter region frequency bands that is actually propagated to the vehicle. A human sensation weight-setting unit **354** sets weights for the vibration from the float region, bounce region, and flutter region frequency bands that is actually propagated to a passenger.

The setting of human sensation weights will now be described. FIG. **14** is a correlation graph showing human sense profiles plotted against frequency. As shown in FIG. **14**, passenger sensitivity to frequencies is comparatively low in the low-frequency float region, with sensitivity gradually increasing as one moves into regions of higher frequencies. Frequencies in the flutter region and higher-frequency regions become progressively harder to transmit to the passenger. In view of this, the float region human sensation weight Wf is set at 0.17, the bounce region human sensation weight Wh is set higher than Wf at 0.34, and the flutter region human sensation weight Wb is set higher than Wf and Wh at 0.38. It is thereby possible to increase the correlation between the scalar quantities for the various frequency bands and the vibration actually propagated to a passenger. These two weighting factors may be altered, as appropriate, according to vehicle concept or passenger preferences.

A weight-determining device **355** calculates the proportions occupied by the weight for each of the frequency bands. Defining a as the float region weight, b as the bounce region weight, and c as the flutter region weight, the weighting factor for the float region is (a/(a+b+c)), the weighting factor for the bounce region is (b/(a+b+c)), and the weighting factor for the flutter region is (c/(a+b+c)).

A scalar quantity calculator unit **356** multiplies the scalar quantities for the various frequency bands calculated by the Hilbert transform processor unit **352** by the device **355**, and outputs final scalar quantities. The process up to this point is performed on the wheel speed sensor values for each of the wheels.

A maximum value-selecting unit **357** selects the maximum value out of the final scalar quantities calculated for each of the four wheels. The 0.01 at the bottom is set to avoid having 0 as a denominator, as the total of the maximum values is used as a denominator in a subsequent process. A proportion calculator unit **358** calculates proportions using the total of the maximum scalar quantity values for each of the frequency bands as the denominator and the maximum scalar quantity value for the frequency band equivalent to a float region as the numerator. In other words, the proportion of contamination (hereafter, simply "proportion") in the float region contained in all vibration components is calculated. A sprung mass resonance filter **359** performs a filter process having a sprung mass resonance frequency of roughly 1.2 Hz on the calculated proportion, and extracts a sprung mass resonance frequency band component representing the float region from the cal-

culated proportion. In other words, because the float region is present at roughly 1.2 Hz, it is believed that the proportion of this region will also vary around 1.2 Hz. The finally extracted proportion is then outputted to the damping force control unit **35**, and a frequency-sensitive damping force control amount corresponding to the proportion is outputted.

FIG. **15** is a plot showing the relationship between the proportion of vibration contamination and damping force in a float region in the frequency-sensitive control of the first embodiment. As shown in FIG. **15**, a high damping force is set when the float region occupies a large proportion, thereby reducing the vibration level of sprung mass resonance. Even if a high damping force is set, no high-frequency vibration or bounce-like vibration is transmitted to the passenger because the bounce region and flutter region occupy small proportions. Meanwhile, setting a low damping force when the float region occupies a small proportion reduces the vibration transmission profile at and above sprung mass resonance, minimizing high-frequency vibration and yielding a smooth ride.

The benefits of frequency-sensitive control in a comparison of frequency-sensitive control and skyhook control will now be described. FIG. **16** is a wheel speed frequency profile detected by a wheel speed sensor **5** in certain driving conditions. This profile especially appears when driving on road surfaces having continuous small bumps, such as cobbled roads. When skyhook control is performed while driving on a road surface exhibiting this profile, the problem arises that, because damping force is determined using the peak amplitude value in skyhook control, any degradation in phase estimation for high-frequency vibrational input will cause an extremely high damping force to be set at the wrong timing, leading to exacerbated high-frequency vibration.

By contrast, if control is performed using scalar quantities rather than vectors, as in frequency-sensitive control, the float region occupies a small proportion on road surfaces such as that shown in FIG. **16**, leading to a low damping force being set. Thus, even if the amplitude of flutter region vibration is high, the vibration transmission profile is sufficiently reduced, allowing the exacerbation of high-frequency vibration to be avoided. As shown by the foregoing, high-frequency vibration can be minimized via scalar quantity-based frequency-sensitive control in regions where control is difficult due to degradations in phase estimation precision even if skyhook control is performed using an expensive sensor.

(Unsprung Mass Vibration Damping Control Unit)

Next, the configuration of the unsprung mass vibration damping control unit will be described. As discussed in the context of the conventional vehicle shown in FIG. **9**B, a resonance frequency band is also present in tires, as they possess both a modulus of elasticity and a damping coefficient. However, because a tire has a mass that is less than that of the sprung mass, and a high modulus of elasticity as well, the band is present toward frequencies higher than sprung mass resonance. The unsprung mass resonance component causes tire rumbling in the unsprung mass, potentially degrading ground contact. In addition, rumbling in the unsprung mass can be uncomfortable for passengers. Thus, damping force is set according to the unsprung mass resonance component in order to minimize unsprung mass resonance-induced rumbling.

FIG. **17** is a block diagram showing a control configuration for unsprung mass vibration damping control in the first embodiment. An unsprung mass resonance component extractor unit **341** applies a band-pass filter to the wheel speed variation outputted from the deviation calculator unit **321***f* of the driving state estimator unit **32** to extract an unsprung mass resonance component. The unsprung mass resonance component is extracted from the region at roughly 10-20 Hz in the wheel speed frequency component. An envelope waveform-forming unit **342** scalarizes the extracted unsprung mass resonance component, and forms an envelope waveform using an envelope filter. A gain multiplier unit **343** multiplies the scalarized unsprung mass resonance component by the gain, calculates an unsprung mass vibration damping force control amount, which is outputted to the damping force control unit **35**

(Configuration of Damping Force Control Unit)

Next, the configuration of the damping force control unit **35** will be described. FIG. **18** is a control block diagram showing a control configuration for a damping force control unit of the first embodiment. The driver input damping force control amount outputted from the driver input control unit **31**, the S/A orientation control amount outputted from the skyhook control unit **33***a*, the frequency-sensitive damping force control amount outputted from the frequency-sensitive control unit **33***b*, the unsprung mass vibration damping force control amount outputted from the unsprung mass vibration damping control unit **34**, and the stroke speed calculated by the driving state estimator unit **32** are inputted into an equivalent viscous damping coefficient converter unit **35***a*, which converts these values into an equivalent viscous damping coefficient.

A damping coefficient-reconciling unit **35***b* reconciles which damping coefficient, out of the damping coefficients converted by the equivalent viscous damping coefficient converter unit **35***a* (hereafter referred to individually as the driver input damping coefficient k1, the S/A orientation damping coefficient k2, the frequency-sensitive damping coefficient k3, and the unsprung mass vibration damping coefficient k4), is used to perform control, and outputs a final damping coefficient. A control signal converter unit **35***c* converts a control signal (commanded current value) to be sent to the S/As **3** on the basis of the damping coefficient reconciled by the damping coefficient-reconciling unit **35***b* and the stroke speed, and outputs the signal to the S/As **3**.

(Damping Coefficient-Reconciling Unit)

Next, the specifics of the reconciliation performed by the damping coefficient-reconciling unit **35***b* will be described. The vehicle control device of the first embodiment has four control modes. The first mode is standard mode, for situations in which suitable steering conditions are obtainable while driving on general urban roads. The second mode is sports mode, for situations in which stable steering conditions are available while aggressively driving along winding roads and the like. The third mode is comfort mode, for situations in which priority is given to comfort while driving, such as when starting off at low vehicle speeds. The fourth mode is highway mode, for situations involving driving at high vehicle speeds on highways and the like with multiple straight sections.

In standard mode, priority is given to unsprung mass vibration damping control performed by the unsprung mass vibration damping control unit **34** while skyhook control is being performed by the skyhook control unit **33***a*. In sports mode, skyhook control is performed by the skyhook control unit **33***a* and unsprung mass vibration damping control is performed by the unsprung mass vibration damping control unit **34** while giving priority to driver input control performed by the driver input control unit **31**. In comfort mode, priority is given to unsprung mass vibration damping control performed by the unsprung mass vibration damping control unit **34** while frequency-sensitive control is being performed by the frequency-sensitive control unit **33***b*. In highway mode, the control amount for the unsprung mass vibration damping control

performed by the unsprung mass vibration damping control unit **34** is added to the skyhook control performed by the skyhook control unit **33***a* while given priority to the driver input control performed by the driver input control unit **31**. Damping coefficient reconciliation in these various modes will now be described.

(Reconciliation in Standard Mode)

FIG. **19** is a flow chart of a damping coefficient reconciliation process performed during a standard mode in the first embodiment.

In step S**1**, it is determined whether the S/A orientation damping coefficient k2 is greater than the unsprung mass vibration damping coefficient k4, and, if so, the process continues to step S**4**, and k2 is set as the damping coefficient.

In step S**2**, the scalar quantity proportion of the flutter region is calculated on the basis of the scalar quantities for the float region, bounce region, and flutter region described in the context of the frequency-sensitive control unit **33***b*.

In step S**3**, it is determined whether the proportion of the flutter region is equal to or greater than a predetermined value, and, if so, the process continues to step S**4** and the low value k2 is set as the damping coefficient for fear of high-frequency vibration reducing ride comfort. On the other hand, if the proportion of the flutter region is less than the predetermined value, there is no worry of high-frequency vibration reducing ride comfort even if a high damping coefficient is set, so the process continues to step S**5**, and k4 is set as the coefficient.

In standard mode, as discussed above, priority is given, as a rule, to unsprung mass vibration damping control, which minimizes resonance in the unsprung mass. However, if the damping force required for skyhook control is less than the damping force required for unsprung mass vibration damping control, and the flutter region occupies a large proportion, the damping force for skyhook control is set so as to avoid exacerbating the high-frequency vibration profile in order to meet the requirements of unsprung mass vibration damping control. This allows an optimal damping profile to be obtained according to the driving state, allowing high-frequency vibration-induced degradations of ride comfort to be avoided while simultaneously achieving a flat vehicle body feel.

(Reconciliation in Sports Mode)

FIG. **20** is a flow chart of a damping coefficient reconciliation process performed during a sports mode in the first embodiment.

In step S**11**, the damping force distribution factors for the four wheels are calculated on the basis of the driver input damping coefficients k1 for the four wheels set during driver input control. Defining k1fr as the front right wheel driver input damping coefficient, k1fl as the front left wheel driver input damping coefficient, kin as the rear right wheel driver input damping coefficient, k1rl as the rear left wheel driver input damping coefficient, and xfr, xfl, xrr, and xrl as the damping force distribution factors for the different wheels, the distribution factors are calculated as follows:

$$xfr=k1fr/(k1fr+k1fl+k1rr+k1rl)$$

$$xfl=k1fl/(k1fr+k1fl+k1rr+k1rl)$$

$$xrr=k1rr/(k1fr+k1fl+k1rr+k1rl)$$

$$xrl=k1rl/(k1fr+k1fl+k1rr+k1rl)$$

In step S**12**, it is determined whether a damping force distribution factor x is within a predetermined range (greater than α and less than β), and, if so, distribution is determined to be roughly equal for all the wheels, and the process continues to step S**13**; if even one factor is outside the predetermined range, the process continues to step S**16**.

In step S**13**, it is determined whether the unsprung mass vibration damping coefficient k4 is greater than the driver input damping coefficient k1, and, if so, the process continues to step S**15**, and k4 is set as a first damping coefficient k. On the other hand, if the unsprung mass vibration damping coefficient k4 is equal to or less than the driver input damping coefficient k1, the process continues to step S**14**, and k1 is set as the first damping coefficient k.

In step S**16**, it is determined whether the unsprung mass vibration damping coefficient k4 is the maximum value max that can be set for the S/As **3**; if so, the process continues to step S**17**, and, if not, the process continues to step S**18**.

In step S**17**, the maximum value for the driver input damping coefficients k1 for the four wheels is the unsprung mass vibration damping coefficient k4, and the damping coefficient that satisfies the damping force distribution factor is calculated as the first damping coefficient k. In other words, a value is calculated such that the damping coefficient is maximized while the damping force distribution factor is satisfied.

In step S**18**, a damping coefficient such that the damping force distribution factor is satisfied within a range in which the driver input damping coefficients k1 for all four wheels are equal to or greater than k4. In other words, a value is calculated such that the damping force distribution factor set by the driver input control is satisfied, and the requirements of unsprung mass vibration damping control are also met.

In step S**19**, it is determined whether the first damping coefficients k set in the abovementioned steps are less than the S/A orientation damping coefficient k2 set during skyhook control; if so, k2 is set and the process continues to step S**20** due to the damping coefficient required by skyhook control being larger. On the other hand, if k is equal to or greater than k2, k is set and the process continues to step S**21**.

In sports mode, as discussed above, priority is given, as a rule, to unsprung mass vibration damping control, which minimizes resonance in the unsprung mass. However, because the damping force distribution factor required by driver input control is closely related to the vehicle body orientation, and is particularly deeply related to changes in driver line of view caused by roll mode, foremost priority is given to ensuring the damping force distribution factor, rather than the damping coefficient required by driver input control itself. For movement that causes changes in vehicle body orientation while preserving the damping force distribution factor, selecting skyhook control via select high allows a stable vehicle body orientation to be maintained.

(Reconciliation in Comfort Mode)

FIG. **21** is a flow chart of a damping coefficient reconciliation process performed during a comfort mode in the first embodiment.

In step S**30**, it is determined whether the frequency-sensitive damping coefficient k3 is greater than the unsprung mass vibration damping coefficient k4, and, if so, the process continues to step S**32** and the frequency-sensitive damping coefficient k3 is set. On the other hand, if the frequency-sensitive damping coefficient k3 is determined to be equal to or less than the unsprung mass vibration damping coefficient k4, the process continues to step S**32** and the unsprung mass vibration damping coefficient k4 is set.

In comfort mode, as discussed above, priority is given, as a rule, to unsprung mass resonance damping control, which minimizes resonance in the unsprung mass. Because frequency-sensitive control is performed as sprung mass vibration damping control to begin with, and an optimal damping coefficient for the road surface conditions is set, control can

be performed while ensuring ride comfort, allowing sensations insufficient ground contact caused by rattling in the unsprung mass to be avoided through unsprung mass vibration damping control. In comfort mode, as in standard mode, it is acceptable for the damping coefficient to be switched according to the proportion of flutter in the frequency scalar quantity. This allows for a super comfort mode in which ride comfort is even better ensured.

(Reconciliation in Highway Mode)

FIG. **22** is a flow chart of a damping coefficient reconciliation process performed during a highway mode in the first embodiment. The same reconciliation process as in sports mode is performed from steps S**11** to S**18**; thus, description thereof will be omitted.

In step S**40**, the S/A orientation damping coefficient k2 yielded by skyhook control is added to the reconciled first damping coefficient k yielded by the process up to step S**18** and outputted.

In highway mode, as discussed above, the sum of the reconciled first damping coefficient k and the S/A orientation damping coefficient k2 is used to reconcile the damping coefficient. This operation will now be described with reference to the drawings. FIG. **23** is a time chart showing changes in damping coefficient when driving on a hilly road surface and a bumpy road surface. For instance, if an attempt is made to minimize swaying motion in the vehicle body caused by the effects of slight hills in the road surface when driving at high vehicle speeds via skyhook control alone, it is necessary to detect slight variations in wheel speed, which requires that a comparatively high skyhook control gain be set. In such cases, swaying motion can be minimized, but bumps in the road surface can lead to the control gain being too great, creating the risk of excessive damping force control being performed. This gives rise to concerns of degraded ride comfort or vehicle body orientation.

By contrast, because the first damping coefficient k is constantly set, as in highway mode, a certain level of damping force can be constantly ensured, allowing swaying motion in the vehicle body to be minimized even if a low damping coefficient is used in skyhook control. In addition, because there is no need to increase the skyhook control gain, bumps in the road surface can be managed using a normal control gain. Moreover, because skyhook control is performed in a state in which the damping coefficient k is set, a process of reducing the damping coefficient can be operated in a semi-active control region, unlike in the case of a damping coefficient limit, ensuring a stable vehicle orientation during high-speed driving.

(Mode Selection Process)

Next, a mode selection process for selecting among the various driving modes described above will be described. FIG. **24** is a flow chart of a driving state-based mode selection process performed by a damping coefficient-reconciling unit of the first embodiment.

In step S**50**, it is determined whether the vehicle is driving straight ahead based on the value from the steering angle sensor **7**; if so, the process continues to step S**51**, and if the vehicle is determined to be in a state of turning, the process continues to step S**54**.

In step S**51**, it is determined whether the vehicle speed is equal to or greater than a predetermined vehicle speed VSP1 indicating a state of high vehicle speed on the basis of the value from the vehicle speed sensor **8**, and, if so, the process continues to step S**52** and standard mode is selected. On the other hand, if the speed is less than VSP1, the process continues to step S**53** and comfort mode is selected.

In step S**54**, it is determined whether the vehicle speed is equal to or greater than a predetermined vehicle speed VSP1 indicating a state of high vehicle speed on the basis of the value from the vehicle speed sensor **8**, and, if so, the process continues to step S**55** and highway mode is selected. On the other hand, if the speed is less than VSP1, the process continues to step S**56** and sports mode is selected.

That is, standard mode is selected when driving at a high vehicle speed when driving straight ahead, thereby making it possible to stabilize the vehicle body orientation via skyhook control, ensure ride comfort by minimizing high-frequency vibration-induced bouncing or fluttering, and minimizing resonance in the unsprung mass. Selecting comfort mode when driving at low speeds makes it possible to minimize resonance in the unsprung mass while minimizing the transmission of vibration such as bouncing or fluttering to passengers.

Meanwhile, highway mode is selected when driving at a high vehicle speed in a state of turning, thereby performing control using a value to which a damping coefficient has been added; thus, high damping force is yielded as a rule. It is thus possible to minimize unsprung mass resonance while actively ensuring the unsprung mass resonance during turning via driver input control, even when traveling at a high vehicle speed. Selecting sports mode when driving at a low vehicle speed allows unsprung mass resonance to be minimized while actively ensuring the vehicle body orientation during turning via driver input control and performing skyhook control as appropriate, thereby allowing for driving with a stable vehicle orientation.

In the first embodiment, an example of a mode selection process in which the driving state is detected and the mode is automatically switched has been presented, but it is also possible to provide a mode switch or the like that can be operated by a driver to select the driving mode. This yields ride comfort and turning performance matching the driver's desired driving state.

As described above, the first embodiment yields the following effects.

(1) Provided are: S/As **3** (variable-damping-force shock absorbers) of adjustable damping force; front wheel speed sensors **5**FL, **5**FR, connected to a front wheel axle, for detecting front wheel speed; wheel speed sensors **5**RL, **5**RR for detecting rear wheel speed on the basis of a rear drive shaft rotational rate outputted by a rear wheel differential gear; a sprung rear wheel speed estimator unit **305***a* and an unsprung rear wheel speed estimator unit **305***b* (a rear wheel speed estimation device) for delaying the front wheel speed detected by the front wheel speed sensors **5**FL, **5**FR and estimating rear wheel speed; a damping force control amount calculation device for taking the front wheel speed detected by the front wheel speed sensors **5**FL, **5**FR and the size of the amplitude of the frequency of the rear wheel speed estimated by the sprung rear wheel speed estimator unit **305***a* as frequency scalar quantities, and calculating a damping force control amount for the S/As **3** according to the frequency scalar quantities; and a damping force control unit **35** (a damping force control device) for controlling the damping force of the S/As **3** on the basis of the damping force control amount calculated by the damping force control amount calculation device.

Because there is little change in the values detected by the rear wheel speed sensors **5**RL, **5**RR in response to suspension stroke-induced axle tilting, and such axle tilting is not readily reflected in changes in the wheel speed sensor values, using the rear wheel speed detected by the rear wheel speed sensors **5**RL, **5**RR will reduce stroke speed estimation precision,

potentially negatively affecting vibration damping performance. On the other hand, the wheel speed sensors **5**FL, **5**FR for the front wheels connected to the axle are capable of precisely estimating stroke speed due to the large changes in wheel speed sensor values that occur in response to stroke-induced axle tilting. As a result, the front wheel speed sensor value is delayed to estimate the rear wheel speed, thereby improving stroke speed estimation precision and allowing reductions in vibration damping performance to be minimized.

Because the damping force control amount for the S/As **3** is calculated on the basis of frequency scalar quantities, there is no reduction in vibration damping performance even if there is a deviation in phase between the estimated rear wheel speed and the actual rear wheel speed, allowing vibration damping performance to be improved using a precisely estimated stroke speed.

(2) The frequency-sensitive control unit **33***b* (a damping force control amount calculation device) calculates a frequency-sensitive damping force control amount for the S/As **3** according to the proportion of a predetermined frequency region of the wheel speed sensor value occupied by a flutter region.

In vector control in which the sign relationship (phase, etc.) between the stroke speed and the sprung mass speed is vital, as in the case of skyhook control, slight deviations in phase may impede appropriate control being performed, and if the front wheel speed sensor value is delayed to estimate the rear wheel speed, the sign relationship between the stroke speed and the sprung mass speed cannot be guaranteed, potentially reducing vibration damping performance. By contrast, frequency-sensitive control, which is sprung vibration damping control commensurate with scalar quantities in the vibration profile, is not dependent upon the sign relationship between the stroke speed and the sprung mass speed; i.e., the control amount is determined regardless of the signs. Thus, a rear wheel speed estimated by delaying the front wheel speed sensor value can be used to improve vibration damping performance.

(3) The unsprung vibration damping control unit **34** (a damping force control amount calculation device) calculates the unsprung vibration damping force control amount on the basis of the amplitude of the frequency of the unsprung resonance.

The unsprung vibration damping force control amount is calculated by extracting an unsprung resonance component from changes in wheel speed, and multiplying a scalarized value for the component by gain. In other words, there is no dependency upon the sign relationship between the stroke speed and the sprung mass speed, and the control amount is determined regardless of the signs, allowing a rear wheel speed estimated by delaying the front wheel speed sensor values to be used to improve vibration damping performance.

(4) The damping force control amount calculation device generates a frequency envelope amplitude, and calculates frequency scalar quantities on the basis of the envelope amplitude.

Specifically, in frequency-sensitive control, a Hilbert transform processor unit **352** performs a Hilbert transform upon various frequency bands divided into a float region, a bounce region, and a flutter region, and converts to scalar quantities based on the amplitudes of the frequencies. Meanwhile, the unsprung vibration damping control unit **34** uses an envelope waveform-generating unit **342** to scalarize an unsprung resonance component extracted by an unsprung resonance component-extracting unit **341**, and uses an envelope filter to generate an envelope waveform.

It is thus possible to calculate frequency scalar quantities using a simple method.

(5) When the vehicle speed is equal to or greater than the high vehicle speed threshold value V2, the rear wheel speed estimation device prohibits rear wheel speed estimation.

Specifically, if the vehicle speed is equal to or greater than the high vehicle speed threshold value V2, the front wheel first wheel speed is not stored in memory, and the rear wheel first wheel speed inputted from the flat surface motion component extractor unit **301** after the delay time has elapsed is outputted, thereby preventing an impossible delay time from being set and control performance from being reduced.

(6) When the vehicle speed is less than the low vehicle speed threshold value V1, the rear wheel speed estimation device prohibits rear wheel speed estimation.

Specifically, if the vehicle speed is less than the low vehicle speed threshold value V1, the front wheel first wheel speed is not stored in memory, and the rear wheel first wheel speed inputted from the flat surface motion component extractor unit **301** after the delay time has elapsed is outputted, thereby preventing program malfunctions arising from buffer overflows.

The invention claimed is:

1. A vehicle control device comprising:

a variable-damping-force shock absorber configured to adjust damping force on front and rear wheels;

a front wheel speed sensor configured to detect front wheel speed, the front wheel speed sensor being configured to connect to a front wheel axle;

a rear wheel speed estimation device configured to delay the front wheel speed detected by the front wheel speed sensors to estimate rear wheel speed;

a damping force control amount calculation device configured to calculate a damping force control amount for the variable-damping-force shock absorber according to frequency scalar quantities, when the front wheel speed detected by the front wheel speed sensors and the size of an amplitude of a frequency of the rear wheel speed estimated by the rear wheel speed estimation device are frequency scalar quantities; and

a damping force control device configured to control the damping force of the variable-damping-force shock absorber on the basis of the damping force control amount calculated by the damping force control amount calculation device;

the rear wheel speed estimation device prohibiting the estimation of the real wheel speed when a vehicle speed is equal to or greater than a high vehicle speed threshold value.

2. The vehicle control device according to claim **1**, wherein the damping force control amount calculation device is configured to calculate the damping force control amount for the variable-damping-force shock absorber according to a ratio of frequency scalar quantities in a first predetermined frequency band and frequency scalar quantities in a second frequency band.

3. The vehicle control device according to claim **2**, wherein the damping force control amount calculation device is configured to calculate a damping force control amount for unsprung-mass vibration damping control on the basis of the amplitude of an unsprung resonance frequency.

4. The vehicle control device according to claim **1**, wherein the damping force control amount calculation device is configured to calculate a damping force control amount

for unsprung-mass vibration damping control on the basis of the amplitude of an unsprung resonance frequency.

5. The vehicle control device according to claim **4**, wherein the damping force control amount calculation device is configured to generate an envelope amplitude for the frequency, and calculates the frequency scalar quantities on the basis of the envelope amplitude.

6. The vehicle control device according to claim **1**, wherein the damping force control amount calculation device is configured to generate an envelope amplitude for the frequency, and calculate the frequency scalar quantities on the basis of the envelope amplitude.

7. The vehicle control device according to claim **6**, wherein the damping force control amount calculation device is configured to generate an envelope amplitude for the frequency, and calculates the frequency scalar quantities on the basis of the envelope amplitude.

8. A vehicle control device comprising:

a variable-damping-force shock absorber configured to adjust damping force on front and rear wheels;

a front wheel speed sensor configured to detect front wheel speed, the front wheel speed sensor being configured to connect to a front wheel axle;

a rear wheel speed estimation device configured to delay the front wheel speed detected by the front wheel speed sensors to estimate rear wheel speed;

a damping force control amount calculation device configured to calculate a damping force control amount for the variable-damping-force shock absorber according to frequency scalar quantities, when the front wheel speed detected by the front wheel speed sensors and the size of an amplitude of a frequency of the rear wheel speed estimated by the rear wheel speed estimation device are frequency scalar quantities; and

a damping force control device configured to control the damping force of the variable-damping-force shock absorber on the basis of the damping force control amount calculated by the damping force control amount calculation device;

the rear wheel speed estimation device prohibiting the estimation of the real wheel speed when vehicle speed is less than a low vehicle speed threshold value.

9. The vehicle control device according to claim **8**, wherein

the damping force control amount calculation device is configured to calculate the damping force control amount for the variable-damping-force shock absorber according to a ratio of frequency scalar quantities in a first predetermined frequency band and frequency scalar quantities in a second frequency band.

10. The vehicle control device according to claim **9**, wherein

the damping force control amount calculation device is configured to calculate a damping force control amount for unsprung-mass vibration damping control on the basis of the amplitude of an unsprung resonance frequency.

11. The vehicle control device according to claim **9**, wherein

the damping force control amount calculation device is configured to generate an envelope amplitude for the frequency, and calculates the frequency scalar quantities on the basis of the envelope amplitude.

12. The vehicle control device according to claim **8**, wherein

the damping force control amount calculation device is configured to calculate a damping force control amount for unsprung-mass vibration damping control on the basis of the amplitude of an unsprung resonance frequency.

13. The vehicle control device according to claim **8**, wherein

the damping force control amount calculation device is configured to generate an envelope amplitude for the frequency, and calculates the frequency scalar quantities on the basis of the envelope amplitude.

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